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IN THIS ISSUE

A RECENT addition to the vocabulary of high-voltage power transmission is the term "group conductor," or "Bündelleiter" as it is called in Germany, where, we have reason to believe, the first large-scale experiments* with this new development were conducted during the last War.

The classic method of preventing or diminishing the corona phenomenon in high-voltage transmission lines is to increase the diameter of the conductors. Although it has been found possible to do this without increasing their cross-sectional area beyond an optimum value dictated by economic considerations, design and erection of such transmission lines are beset with serious limitations and drawbacks, especially if power transmission of the order of 500 kV is envisaged.

A new approach to this problem is the use of two, three or four wires per phase, spaced at appropriate distances apart and each having a smaller diameter than the single large-diameter cable hitherto employed for the same purpose. The resulting gradient of field strength of the conductors can, by this means, be kept small enough to prevent the occurrence of the corona phenomenon, even at very high voltages and under comparatively unfavourable weather conditions. Development work in this field is now in progress in the United States** and also in European countries. The article on page 295 presents a summary of investigations carried out in France. The new system outlined above may well prove to be one of the decisive factors in favour of A.C. transmission in the present controversy between the advocates of A.C. and D.C. for the transmission of large power blocks over great distances.

* * *

In the design of stirring apparatus for viscous liquids, due consideration must be given to the question of eliminating "whipping" effects at operating speeds. The problem is rendered more complicated by the fact that not only the flexibility of the stirrer shaft, but also the elasticity of the supports, has to be taken into account. A simple method of obtaining the results required is outlined in the article "Vibrations of Vertical Vane-Type Stirrers" on page 301 in this Issue.

* * *

It may perhaps appear to be a little unusual to make editorial mention of an article which has only been given a very limited amount of space in our columns. The original paper, a short outline of which is given by the classified abstract "Wear Technology," is really worthy of greater prominence. A comprehensive abstract, however, would have occupied too much of the limited space at our disposal.

The article is valuable mainly because it shows to what extent research has still to be carried out before all questions regarding wear can be answered sufficiently to provide definite guidance towards the solution of practical problems. From the bibliography, which comprises 31 references, it appears that comparatively little work along these lines has recently been done in Great Britain. It is only as regards lubrication and lubricants that some valuable contributions have been made in this country, and these apparently have not received the author's consideration. However, lubrication constitutes only part of the problem, and university and industrial research laboratories may find that the general field of wear technology still affords much scope for their activities.

* * *

Although it is our normal practice to include abstracts from foreign engineering publications only in the pages of our Journal, we feel justified in bringing to our readers' notice a recent publication issued by the Ministry of Supply on "Dimensional Analysis of Engineering Designs," in view of the topical importance of this subject to the present-day production drive.

The publication mentioned is the work of an Inter-Services Committee set up at a Conference held on 24th October, 1944, by the Ministry of Production.

* * *

In France, copper shortage rendered the use of aluminium or aluminium/steel overhead trolley wires† compulsory and, therefore, the problem of designing a suitable aluminium-copper outdoor cable connection was an urgent one, and it was systematically investigated by the Aluminium Research Centre on behalf of the French Railways.

The solution given in the article on page 309 has, of course, a much wider application, especially with reference to modern high-speed isolators, which invariably incorporate aluminium arms with copper contacts in order to reduce the moving masses and yet provide reliable low-resistance contact materials. In connections for overhead wires and high-voltage transmission lines, the same problem has long been recognised to be a very difficult one also in this country, and no agreed standard design has yet been found. The successful outcome of this French development should, therefore, be of particular interest. Should it also prove satisfactory in this country, it would not only eliminate what has until now been a source of irritation and expense, but might also pave the way to a wider adoption of aluminium forgings or even die-castings for outdoor terminals to be used in conjunction with copper cables.

*A report dealing with German developments in this field can be inspected at the Board of Trade, German Division, Technical Information Documents Unit, 40 Cadogan Square, London, S.W.1

**See *The Engineers' Digest*, May, 1948, pp. 169-170.

†See *The Engineers' Digest*, May, 1948, p. 147.

Continued on page 316

SWITZERLAND

The "Hydro-Titan" Infinitely Variable Hydraulic Gear

By T. LUDWIG. (From *Von Roll Mitteilungen*,* Vol. 7, No. 1, March, 1948, pp. 1-12, 13 illustrations.)

INFINITELY variable, positive gears for a wide range of gear ratios and capable of transmitting high powers are of great importance in many branches of engineering. The "Hydro-Titan" gear is based on the well-known principle of axial-piston type oil pumps which transmit the input power hydraulically to a mechanically identical oil motor which transforms the hydraulic energy back into mechanical power. Change of piston stroke of pump and motor effects a control of speed and torque within wide limits. Existing designs of axial- and radial-piston pumps are limited in speed and permissible oil pressure so that comparatively bulky arrangements are required to transmit a given energy. Patented improvements in design have obtained an important reduction in the bulk and weight of the Hydro-Titan gear in comparison with other gears and have facilitated its control. A single gear can transmit several hundred horse-power, with an efficiency of 75 to 85 per cent from half to full load. It can be controlled from rest to maximum speed in both directions, and can be adjusted automatically to given gear ratios in the forward and reverse directions. Gear combinations permit the transmission of high power, and the relatively simple control arrangements can completely eliminate slip and thus allow for a speed control entirely independent of load and temperature variations. Short-time overloads of 100 per cent, and shock loads up to 300 per cent are permissible since the gear is not affected by them. A suitable combination of gears allows for the transmission of high torques at very low speeds not possible with electric drives. The rotating masses are only one tenth the size of those for corresponding electric drives, and the gear can therefore be started, stopped, or reversed rapidly even under full load. It can be driven at any constant speed and the combination of the Hydro-Titan gear with a simple squirrel-cage electro-motor gives a transmission with excellent control possibilities.

The size of the Hydro-Titan gear depends, of course, on the maximum torque to be transmitted. If high and very low speeds of 10 to 50 rpm are required with correspondingly large torques, a fast-running hydraulic gear coupled to toothed reduction gears may be the best arrangement, but if only low speeds are required, a small fast-running oil pump can work in combination with a correspondingly larger oil motor. If the pump has a bore of $\frac{7}{16}$-in, the bore of the motor is $1\frac{1}{4}$-in, then the swept motor volume is 22 times larger than the swept pump volume and a primary coarse reduction of 22 to 1 is thus obtained. Maintenance and servicing of this gear are simple, and all working parts are automatically lubricated with normal machine oil free from air inclusion. The pump and motor need not be mounted as a single unit, but can be separated over distances so long as this does not produce an unreasonable pressure drop in the oil piping. One pump may be used to drive several motors. Fig. 2 shows a planing machine drive with a standard Hydro-Titan gear in place of an electric variable-speed drive. The same control arrangement can be used if the machine is redesigned to have a hydraulic table drive by a Hydro-Titan pump.

Design and Control Improvements Incorporated in Hydro-Titan Gears

An increase in transmitted energy, or a reduction of gear size for a given transmitted power, can only be obtained, for a given swept volume V of the pump, by an increase in fluid pressure or pump speed.

$$N_h = \frac{Q \Delta p}{12 \times 550}\ [\text{hp}] = \frac{p\,Vn}{12 \times 550 \times 60} = \text{const} \times pn$$

where Q is in cub in/sec and p in psi.

The oil pressure can only be increased if the pump construction is robust and rigid (Fig. 3). The oil pressure exerts, via piston and connecting-rod, a radial and axial reaction force on the driving flange which is supported by a half-spherical cradle. This cradle is held by a hollow semi-spherical seating machined in the rigid cast steel housing, having on one side an opening for the driving shaft. On the other side, an axial force

*Published by Louis de Roll Iron Works Ltd., Gerlafingen.

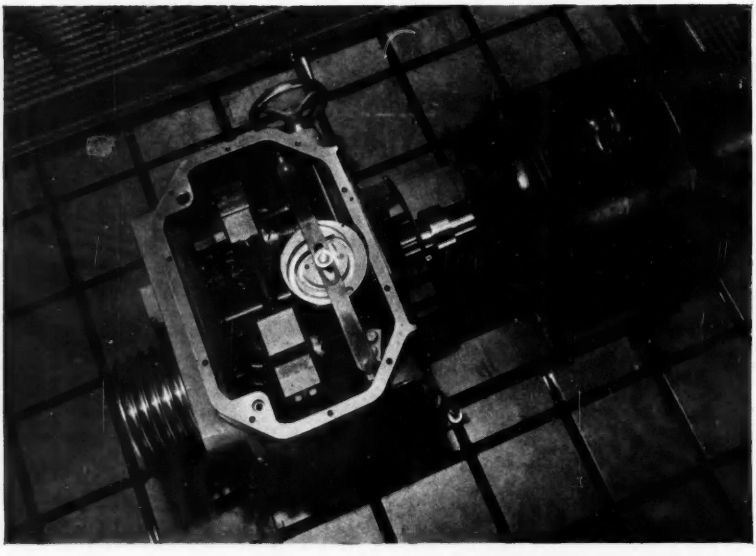


Fig. 1. Type 2532-TL Hydro-Titan gear with cover removed.

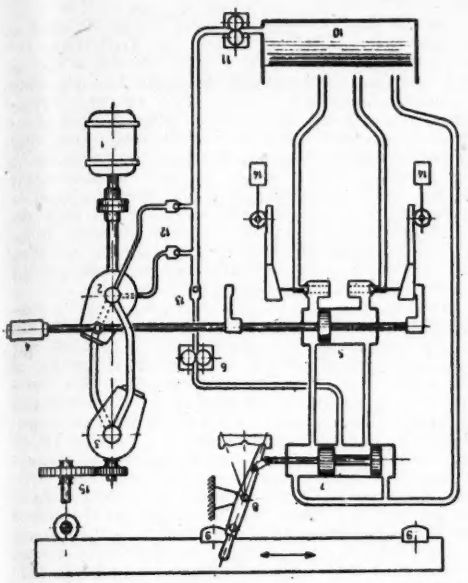


Fig. 2. **Control for the table drive of a planing machine, with a Hydro-Titan gear replacing an electric variable-speed drive.**

(1) Driving motor; (2) Hydro-Titan pump unit; (3) Hydro-Titan motor unit; (4) spring dash-pot; (5) servo-motor; (6) gear pump supplying oil to reversing cylinder; (7) reversing cylinder and pistons; (8) reversing lever; (9) adjustable dogs; (10) oil sump; (11) oil feed pump; (12) in-feed valves for Hydro-Titan unit; (13) non-return valve for reversing circuit; (14) independent adjustment, by hand or servo-motor, of feed and quick-return speeds.

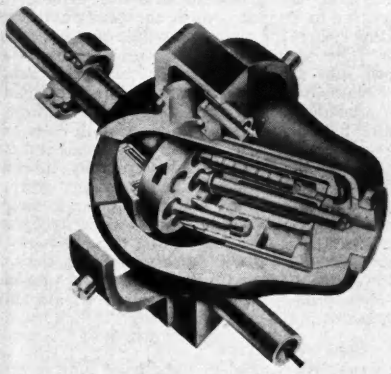


Fig. 3. **High-speed, high-pressure pump unit of Hydro-Titan gear.**

reacts on the drum containing the cylinders and sliding pistons. This drum is positioned at the cylinder head end, by the semi-spherical supporting face of a deep disk conically seated in the cast steel housing. The cradle and the semi-spherical supporting face are partly relieved from the oil reaction pressure by hydraulic arrangements which thus provide for ease of control, and quiet, stable running of the drum, without noticeable wear. This design allowed an appreciable increase in oil pressure and led to surprisingly high efficiencies. An increase in pumping speed at first caused hydrodynamic difficulties which could only be overcome by extensive tests. The high velocities of the oil lead to cavitation and low efficiencies which can only be avoided by pre-compressing the oil by a normal type of gear pump.

Pressure and speed increases give rise to noisy operation and to very high oil velocities through the control slots when oil flows back at the moment of changing the control. A special " pre-compression regulation " effected by the working pressure of the oil, causes a slight tilt of the pump unit in a direction perpendicular to the plane of normal control rotation and this tilt slightly shifts the dead-centre positions of the pistons to counteract, by pre-compression and pre-expansion, the back-flow of the oil. This guarantees quiet operation of the pump even at very high speeds and pressures and has obtained a corresponding reduction in size, as compared with other types of hydraulic gear. The universal suspension of the moving parts, made necessary by these control arrangements, absorbs reaction forces and vibrations in a rigid frame, isolated from the gear box, and no unbalanced forces, nor vibrations, are transmitted to the foundations. Increase in oil speed and pressure, and the reduction of radiating surface areas because of the smaller size of the gear box, introduce the necessity for oil cooling. A certain amount of warm oil is therefore taken out of the low-pressure side of the system, emptied into the sump, pumped back again by the gear feed pump, and led through an oil cooler. Thus, constructional improvements, pressure-oil feed, pre-compression, and oil cooling have contributed to the high power performance of the Hydro-Titan gear.

Characteristics of the Hydro-Titan Gear.

The swept volume of an axial-piston pump is

$$V = z \frac{\pi d^2}{4} D \sin \varphi$$

where d is the bore of a single cylinder, D is the diameter of cylinder pitch circle, z is the number of cylinders, φ is the control angle, i.e. the angle between shaft and cylinder drum axis, which can be assumed to be directly proportional to the swept volume V. The pumped volume of oil can then be written

$$Q = \frac{Vn}{60} \frac{\varphi}{\varphi_{max}} \text{ [cub in/sec]} \quad .. \quad .. \quad (1)$$

where n is the number of strokes of one piston = rpm of pump shaft, V is the maximum total swept pump volume at the maximum control angle φ_{max}, and φ = control angle for a given working position. The transmitted power N, at a given oil pressure p, is

$$N = \frac{p\,Vn}{396000} \frac{\varphi}{\varphi_{max}} \text{ [hp]} \quad .. \quad .. \quad (2)$$

However, since $N = \frac{2\pi n T}{396000}$ [hp], the torque T at the pump shaft is determined by

$$T = \frac{p\,V}{6{\cdot}28} \frac{\varphi}{\varphi_{max}} \text{ [lb in]} \quad .. \quad .. \quad .. \quad (3)$$

These equations are valid for both the pump and motor component of a Hydro-Titan gear. If the oil pump is driven at constant speed n_1 and the oil pressure p is constant, then, according to eq. (2), the transmissible power N will increase as the control angle φ_1 of the pump is increased from zero to maximum, (Fig. 4). An increase in oil pressure would further raise the transmissible power. As the pump control angle φ_1 increases, the pumped oil volume will also increase, see eq. (1), and must be used up by the motor component of the gear. If the motor control angle φ_2 is kept constant, then also according to eq. (1), the motor speed n_2 must increase linearly with the pumped oil

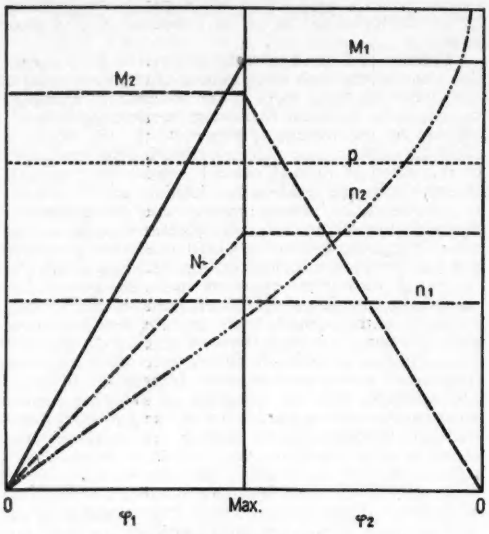


Fig. 4. Characteristic curves of Hydro-Titan gear.

volume. The motor torque T_2, however, remains constant, as can be seen from eq. (3). After the maximum pump control angle $\varphi_{1\,max}$ has been reached, let the pump position remain unchanged, but let, at constant oil pressure p, the motor control angle φ_2 be reduced progressively. Then, since the pumped oil volume Q remains constant, the motor speed n_2 must rise again and the motor torque T_2 must drop according to eqs. (1) and (3) respectively. The oil pressure p does not influence the motor speed but the motor torque T_2 will change linearly with the oil pressure. Since pump and motor control angles can be varied independently or to any desired fixed ratio, the characteristic of the gear is widely adjustable for all purposes.

Design Details. (See assembly drawing, Fig. 5.)

Single units, made up from standard Hydro-Titan high-pressure pumps for presses, are built to suit special requirements, which are mostly for large transmissions of several hundred horse-power. Apart from these special gears, the construction of a standard series is planned which should be controllable within wide limits of speed ratios and oil pressures. Certain special requirements, e.g. regarding the control arrangement, could be satisfied by the provision of non-standard attachments to the standard gear. The following gives a description of the working scheme for a 30 hp Hydro-Titan gear, type TL.

(1) *Arrangement of Units in Gear Box.*

In the TL type gear, the units are arranged side by side in the gear box, with the shafts parallel (Fig. 5). The central driving shaft drives the pump via a reduction gear. The high pressure pipes form a rigid unit with the gear suspension frame which can tilt about horizontal trunnions supported in the side walls of the box. The pump unit can be turned $\pm$ 25 degrees about its vertical axis which is supported in the frame, whilst the motor can only turn between + 8 and + 25 degrees. This gives sufficient control angle latitude for a wide gear ratio and the possibility of reversing the gear through the pump unit. The pre-compression regulation then controls the tilt of $\pm$ 3 degrees maximum about the horizontal axis. The interchangeable reduction gear makes the hydraulic gear independent, within limits, of the speed of the driving electro-motor.

(2) *The Oil Circuit.* (Fig. 6.)

In the pump, mechanical input energy is used to compress oil to a high pressure which oil leaves the pump through a hole in the vertical trunnion at the top (Fig. 3), and is fed through the connecting pipe into the motor unit, where its hydraulic energy is transformed into mechanical output energy. The low-pressure oil leaves the bottom of the motor unit, after expansion, and is led back through the bottom pipe into the pump unit. Certain leakages are unavoidable as, for one reason, the oil is also used to lubricate moving parts. Compression heats the oil with the consequent loss of viscosity, and the efficiency drops. A certain amount of the circulating oil must therefore be taken out continually, cooled and fed back into the circuit, thus replacing losses at the same time.

Oil extraction and in-feed are arranged on the low-pressure side. However, if the gear is reversible, each oil pipe may be the low-pressure pipe so that ball type non-return valves must be arranged at either side of the pump unit. The high-pressure side then automatically closes against the feed pipe. The extraction valves are incorporated in the device for pre-compression regulation near the bottom oil pipe, and a special branch pipe leads from the upper connecting pipe to the pre-compression cylinder and the upper extraction valve. The extraction valve cannot open automatically against the gear pump pressure which acts through the in-feed on to the low-pressure pipe ; therefore, the two extraction valves are connected mechanically by a small push-rod so that the valve on the high-pressure side which is pressed against its seating, opens the valve on the low-pressure side. Oil is thus constantly extracted and flows freely into the sump. The feed pump, a normal gear pump, compresses about 15 per cent of the maximum circulating oil volume to a pressure of 60 to 85 psi. The oil then passes through the cooler in the finned gear box cover, where it is air-cooled from the outside by a fan mounted on the driving shaft, and through the feed pipe to the two in-feed non-return valves. Since in-feed and extraction are arranged for in the same pipe, the gear pump can go on feeding even if the main pump unit does not circulate the oil. The oil fed into the low-pressure pipe simply overflows into the sump through the open extraction valve. No-load or safety valves are not required.

All moving parts and suspension bearings are lubricated by leakage or through special oil holes. The upper suspension bearings, with only small and infrequent movement, need not be specially lubricated. The driving shaft reduction gear and the gear train driving the feed pump are housed in separate compartments of the gear box, and connected only by small oil holes with the main part of the box. At least one gear dips into the oil and splash-lubricates the others and the self-aligning ball bearings. The subdivision of the box prevents the splash oil from introducing air into the main oil bath.

(3) *The Control.*

One handwheel (Fig. 5, right hand bottom sketch) driving a worm gear and a radial cam disk controls both the pump and motor unit. The cam followers rotate both units by means of push bars, and the cam profile is designed to give the most advantageous gear characteristics. Control of the gear is thus very simple and does not require skill. An indicator on the hand wheel shaft shows the direction and speed of rotation of the driven shaft.

(4) *Pre-Compression Regulation.* (Fig. 5, right hand section.)

The pre-compression device is rigidly connected with the lower oil connection pipe on the suspension frame. The high-pressure oil acts on one of the two regulating pistons which are held in the axial position by adjusting screws in the gear box wall. The reaction

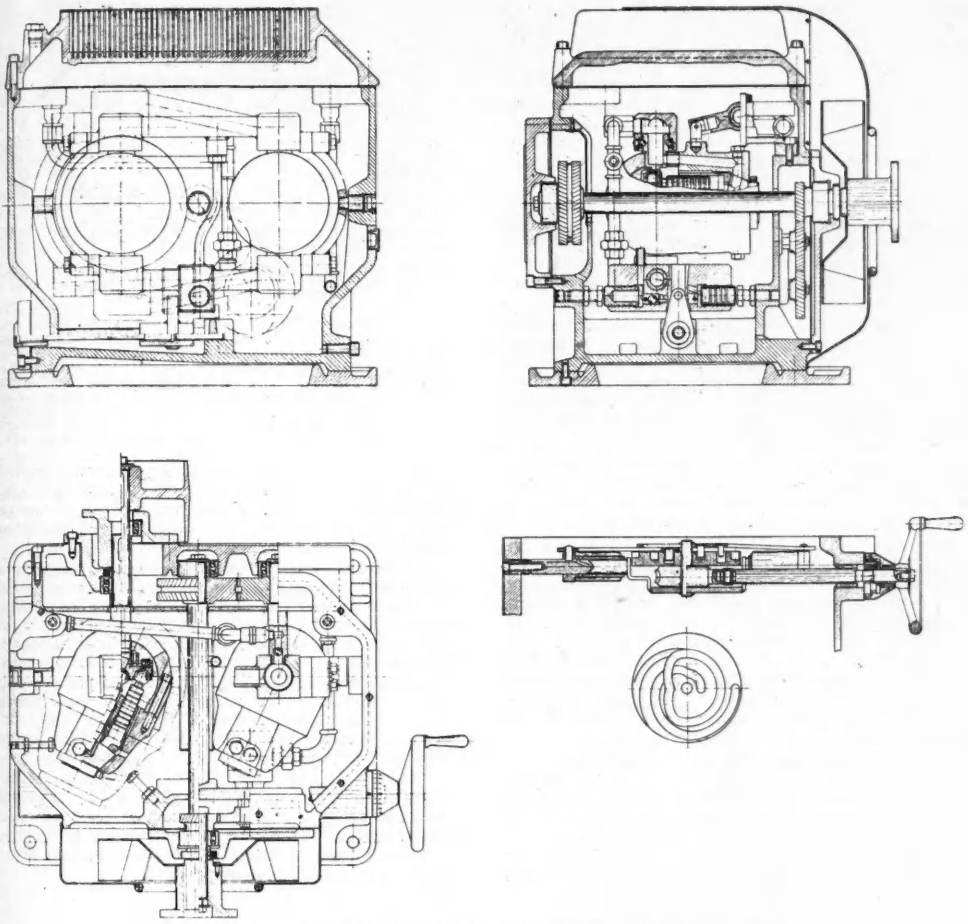


Fig. 5. **Assembly drawing of Hydro-Titan gear type 2525-TL.**

force exerted by the oil on to the cylinder tilts the entire suspension frame about its horizontal axis, overcoming gyroscopic and frictional resistances. The pressure is compensated by a strong torsion spring which ensures that the right amount of tilt is obtained for a given oil pressure. The inactive pre-compression piston on the low-pressure side limits the maximum tilt to 3 degrees.

(5) *The Standard Unit Type TA.* (Figs. 7 and 8.)

This standard Hydro-Titan gear is a tandem arrangement of the two units, with coaxial driving and driven shafts. The external shape and dimensions are about the same as those for an electro-motor of equal power. The gear box casting is ribbed all round and cooled by an external fan on the driving shaft. The common suspension frame for all moving parts is only supported by the shaft bearings in the box; vibrations are thus almost completely suppressed. No special pre-compression regulation is required because the connecting oil pipes can slide axially and their end fittings are held together by tension springs which yield slightly under the oil reaction forces and automatically give the right tilt. The gear box is split axially so that all parts are easily accessible and the gear can be tested and run-in without dismantling the unit or disconnecting the coupling. Otherwise, the working principles are the same as for the TL type gear.

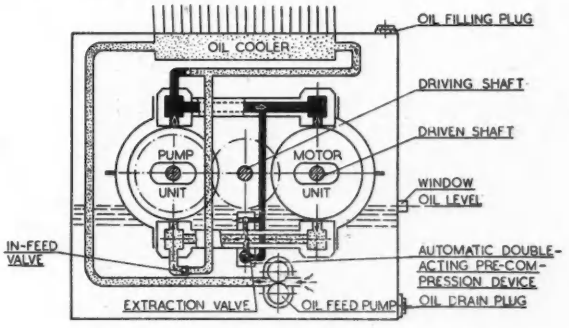


Fig. 6. **Oil circuit of Hydro-Titan gear type TL.**

Possibilities of Application.

Electric drives. Instead of complicated electric variable-speed drives, the simple and reliable three-phase squirrel-cage motor can be used. It can be started under no-load conditions so that even for fairly heavy drives, starters are not required. As long as the rotor inertia is not too high, the squirrel-cage motor can be switched directly on to the mains. Electric reversing drives can also be replaced by the hydraulic gear.

Diesel Locomotives. The drive must possess a wide control range, be efficient to save fuel, guarantee safe functioning and easy operation, and be compact and light in weight. On all these points the Hydro-Titan drive scores over the electric drive. Moreover, it permits smooth starting and great acceleration.

Cars and Lorries. Easy operation, reliability under traffic conditions, and fuel saving efficiency are the main advantages of the Hydro-Titan drive. It replaces the propeller shaft and thus gives greater freedom in arranging coupling, hydraulic gear and differential, which is desirable for the modern, frame-less low-slung vehicle.

Caterpillar Tractors. Each caterpillar is controlled separately, by an arrangement of one pump and two motor units, and one common control lever completely controls the tractor speed and its turning radius.

Machine Tools. Very wide ranges of speed ratios, e.g. milling speeds from 25 to 1000 feet per min., can be obtained with one hydraulic gear. When machining on a lathe, the cutting speed can be kept constant automatically. The Hydro-Titan drive works practically without slip and can therefore even be used as a table drive where very small feeds against high cutting forces are required. Rapid and shock-free reversing of planing machine tables and rolling mill drives is easily obtainable. The drive can be built into existing machine tools.

Starting Difficulties. Machines with large mass inertia need a long starting-up time or, if the starting time is fixed, a high starting torque, perhaps a multiple of the torque required for continuous running. With a Hydro-Titan drive, the engine may start up on no-load and, after it has reached the working speed, the hydraulic gear will accelerate the load with constant, maximum permissible, accelerating torque. Similarly, the Hydro-Titan gear can produce braking torques which are a multiple of the running torque, or can, e.g. in the drive of a heavy centrifuge, feed the braking energy back into the mains.

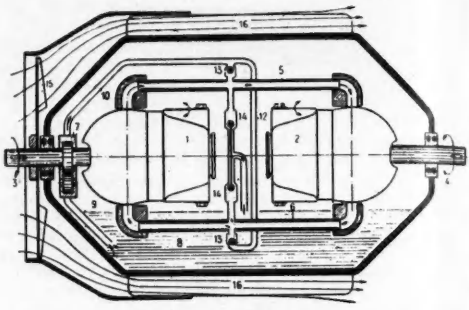


Fig. 7. Hydro-Titan standard gear type TA.
(1) pump unit; (2) motor unit; (3) driving shaft (constant speed and sense of rotation); (4) driven shaft (infinitely variable and reversible); (5) High-pressure oil connecting pipe; (6) Low-pressure oil connecting pipe (on reversal of gear, the high- and low-pressure sides are interchanged); (7) oil feed pump for low-pressure side; (8) oil sump; (9) suction pipe; (10) feed pipe; (12) branch feed pipe; (13) in-feed valves; (14) interconnected extraction valves; (15) fan; (16) cooling fins.

Multiple Drives. The arrangement of one common pump unit driving several motor units coupled directly to different machine shafts is a particularly simple drive. The different shaft speeds or respective angular positions can be brought automatically into the required relationship by the requisite control arrangements. Constant shaft loading (e.g. for constant paper or fibre tension) can easily be secured. The smooth starting and braking characteristics of the Hydro-Titan drive are advantageous where work material has to be carefully handled, e.g. for crane drives in foundries.

Shock Damping. Any shocks from rolling mills or heavy reversing machines or engines appear as pressure waves in the oil circuit and are damped and only partly transmitted to the electric drive. Owing to its shock-absorbing qualities the Hydro-Titan gear can also be used to replace heavy flywheels.

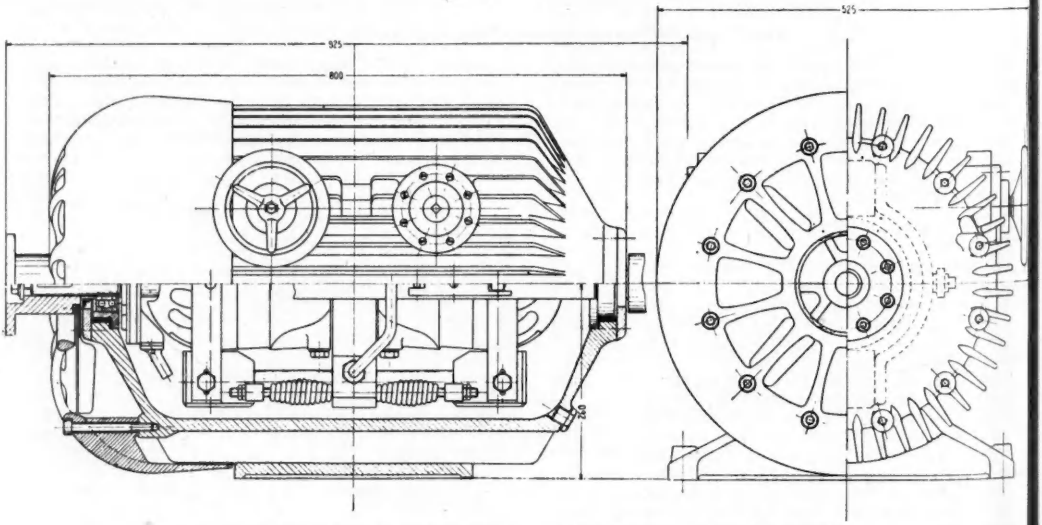


Fig. 8. Elevation and part section of standard Hydro-Titan gear type 2525-TA.

The use of Group Conductors for High-Voltage Transmission Lines

TESTS AT THE 500 kV EXPERIMENTAL STATION AT CHEVILLY

By F. CAHEN and R. PELISSIER. (From *Bulletin de la Société Française des Electriciens*, Series 6, Vol. 8, No. 79, March, 1948, pp. 111-160, 57 illustrations.)

I. INTRODUCTION

THE classic way to prevent or to reduce the occurrence of corona on high voltage transmission lines is to increase the diameter of the conductors without increasing the cross-section beyond the most economical value. Another solution of the problem is to replace the single cable by groups of 2, 3 or 4 conductors having a combined cross-section equal to the most economical cross-section but spaced at a convenient distance from each other.

The effectiveness of this arrangement can be explained with reference to the case of two conductors (Fig. 1). The field in point M on the surface of one conductor is the resultant of a component H_1, produced by the conductor itself, and a much smaller component H_2, produced by the other conductor. H_1, however, has only half the value of the field which would exist if the line consisted of only one single cable of the same radius and was brought up to the same potential. The value of the resulting field varies with the position of point M ; it has a maximum at point A and a minimum at point B ; but at any point of the circumference it is smaller than if the line were equipped with a single conductor of the same radius or with a conductor having a cross-section equal to the combined cross-sections of the two conductors.

Fig. 1a. **Various types of group conductors.**

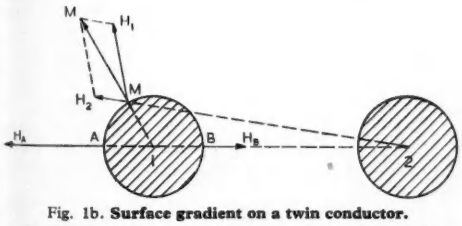


Fig. 1b. **Surface gradient on a twin conductor.**

In France, where the blocks of power transmitted from the South to the North of the country have grown to such an extent that for some time preparations had to be made for the creation of a network to operate at a voltage of the order of 400 kV, special attention was given to the building of double-circuit six-conductor lines with horizontal suspension which could be transformed into single-circuit 400 kV lines.

Such a solution is favourable since it allows for increasing the number of 220 kV lines as and where required, while at the same time leaving open the possibility of changing over rapidly to the higher voltage should that become opportune, a process which, at the same time, would eliminate a number of 220 kV lines.

The first 220 kV transmission line designed for ultimate transformation into a 400 kV line was opened in December, 1946. The line is 405 km long and the transmission towers were designed to carry, ultimately, one 400 kV circuit, either in the form of three hollow

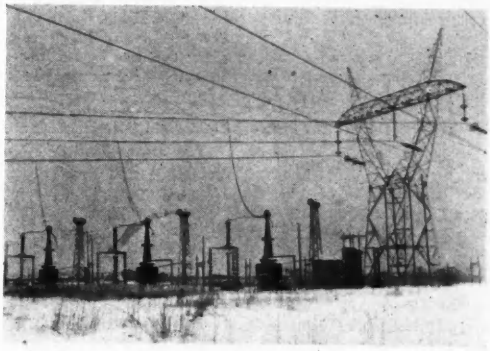


Fig. 2. **General view of the experimental station.**

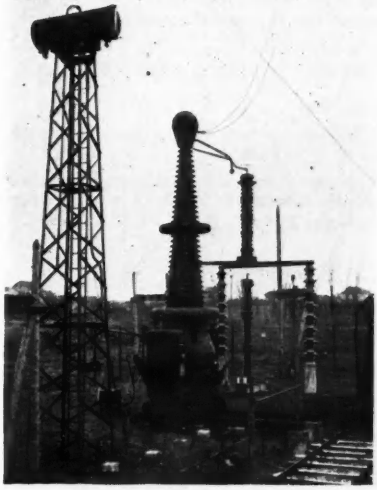


Fig. 3. **Equipment for one phase.**

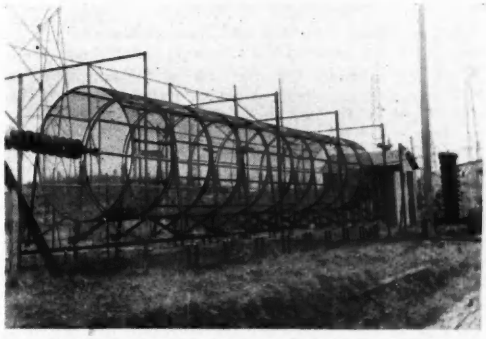


Fig. 4. **Test cage during test of a hollow conductor.**

conductors of large diameter, or in the form of three twin-conductors. The second solution would have the great advantage that changing over to the higher operating voltage would involve merely the regrouping of the existing conductors. As, however, no practical data concerning the lines were available, an experimental station was erected at Chevilly (Figs. 2, 3 and 4) and large-scale experiments and tests were carried out, some of which are described in the following paragraphs which also contain the corresponding theoretical explanations. The group conductors installed and investigated consisted of two conductors per phase and, unless stated otherwise, any reference to a group conductor applies to a twin-conductor.

II. THE POTENTIAL GRADIENT

As the potential gradient at the surfaces of the conductors is the deciding electric factor regarding the formation of corona, a detailed theoretical investigation was carried out on this particular item, so far as the three-phase twin-conductor transmission is concerned. In all calculations the conductor surfaces were assumed to be completely smooth.

(a) *Variation of the Potential Gradient over the surface of the Conductors.* The variation of this gradient has already been briefly mentioned in the introduction, and the following is a very short summary of the detailed calculation. It is assumed that the two conductors are displaced in a horizontal plane and that both are at a potential V_0 to earth. (Fig. 5).

The gradient at point M equals:

$$E = E_m \left(1 + \frac{2r}{D}\cos\alpha\right); \text{ where the mean gradient } E_m = \frac{V_0\sqrt{rD}}{4r\,L_n\,h}; \qquad c = \frac{\sqrt{rD}}{4L_n\,h}$$

is the line capacity of the total group to earth. It is equal to the capacity to earth of a single conductor having a radius $R = \sqrt{r\,D}$.

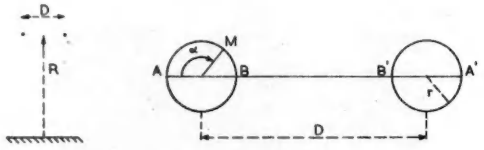


Fig. 5. **Twin-conductor.**

It follows from the above that the gradient varies as a sin function, the maximum and minimum values being:—

$$E_{max} = E_m\left(1 + \frac{2r}{D}\right), \text{ and } E_{min} = E_m\left(1 - \frac{2r}{D}\right)$$

If, for example, the diameter of the conductor is 26·4 mm and the distance between the centres of the conductors is 40 cm, the maximum variation of the gradient is $\pm$ 6 per cent.

(b) *Variation of the Potential Gradient as a function of the Distance between Conductors.* When the distance between the conductors is increased from the position where the two conductors touch, the minimum gradient, as well as the mean gradient, constantly increases. On the other hand, the maximum gradient at first decreases, and then increases after having reached a minimum value $E_{m\ min} = 2V_0/D_m$.

This does not mean, of course, that at this particular distance of the conductors the corona losses are at a minimum, as the total value of the losses is a function of the various gradients over the whole of the surface. However, as the losses increase more rapidly than the respective gradients, it was thought probable that the influence of the zone of the surface with the highest potential gradient would be predominant, and this has been confirmed by experiments as will be shown later.

The calculation of the experimental line shows that the minimum value of the maximum gradient occurs when the distance between the two conductors is 29 cm, while the losses are lowest with a distance of 20 cm. Over 40 cm, the losses as well as the gradient, increase with the distance.

(c) *Disposition of the Conductors of one phase in a Vertical or Horizontal Plane.* As the distance between two conductors is small, compared with the distance from the ground and from the other phases, calculations show that the difference between the vertical and horizontal arrangements is negligible. This has also been confirmed by experiments.

(d) *Single-Phase and Three-Phase-Operation.* In the foregoing only the case of a single-phase group conductor has been considered. In the case of a three-phase transmission line the influence of the other two phases has to be taken into account as well as the influence of the earth.

Detailed calculations were carried out for a three-phase line which included the following alternatives:—

Three-phase operation.

Single-phase operation between one phase and earth, the other two phases being earthed.

Two-phase operation between the two outer phases, the central phase being earthed.

These calculations are based on an assessment of the various capacities. It can be said that in a system of three conductors, the individual charges can be expressed as functions of their potentials, thus:—

$$Q_1 = C_{11}\,V_1 + C_{12}\,V_2 + C_{31}\,V_3.$$
$$Q_2 = C_{12}\,V_1 + C_{22}\,V_2 + C_{23}\,V_3.$$
$$Q_3 = C_{31}\,V_1 + C_{23}\,V_2 + C_{33}\,V_3.$$

Considering, for example, phase 1 and writing the potentials V_2 and V_3 in the form:—

$$V_2 = a_{12}\,V_1,\ V_3 = a_{13}\,V_3,$$

the first equation can be transformed into :—

$$Q_1 = (C_{11} + a_{12}\,C_{12} + a_{13}\,C_{31})\,V_1,$$

or by putting

$$c_1 = C_{11} + a_{12}\,C_{12} + a_{13}\,C_{21},$$
$$Q_1 = c_1\,V_1$$

where c_1 is the apparent capacity of the conductor, taking account of the presence of the two other phases and of the earth and V_1 is the voltage between the conductor and earth.

The potential gradient of a conductor of radius R is

$$E_1 = 2\,Q_1 / R = 2c_1\,V_1 / R,$$

Q_1 and c_1 referring to unit length.

In the case of a group consisting of two conductors both having a radius r, the charge of each conductor is one half of the total charge and the mean gradient over the surface is

$$E_1 = \frac{c_1\ V_1}{r}$$

It is convenient to relate this gradient to the applied voltage and to define a " relative " gradient:—

$$g = \frac{E}{V} = \frac{c}{r}\ [\text{cm}^{-1}]$$

Detailed formulae have been developed for the capacity of a line with horizontal disposition of conductors and for various experimental operating conditions; these formulae have been applied to the calcula-

tion of the capacities and gradients of the experimental line. The results are plotted in Figs. 6 and 7 which show that, especially in three-phase operation, the gradient is higher on the central phase than on the two lateral phases (5 — 7 per cent), that for any of the phases the gradient is higher in three-phase operation than in single- phase operation (8 to 14 per cent), and that it is slightly higher (2 per cent) in two-phase operation than in single-phase operation.

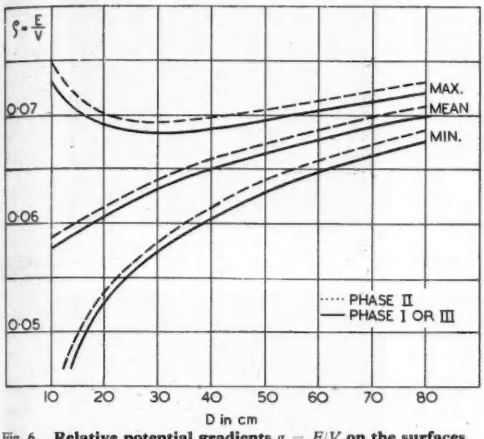


Fig. 6. **Relative potential gradients $g = E/V$ on the surfaces of a twin conductor in single-phase operation as a function of the distance between the two conductors.**

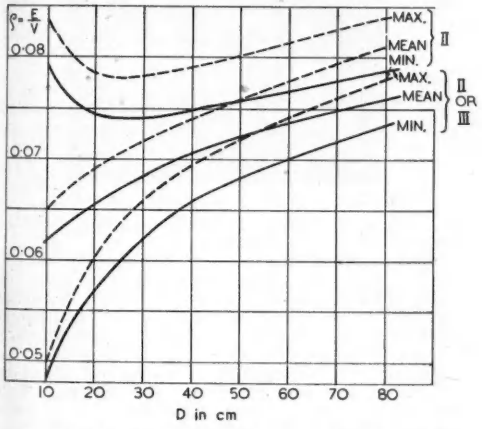


Fig. 7. **Relative potential gradients $g = E/V$ on the surfaces of a twin-conductor in three-phase operation as a function of the distance between the two conductors**

(e) *Influence of the Ground Wire.* The presence of ground wires slightly increases the capacity coefficients, and consequently the surface gradients, and therefore also the corona losses.

(f) *Tests in a Cage.* The calculation of the variation of the potential gradients for the conductors of a group conductor running through a cylindrical cage of 96 cm radius shows that the gradients are identical with those of the real line, provided the voltages applied are inversely proportional to the respective capacities. As the line capacity of the cage is approximately double that of the line, it is sufficient to use, for experiments in the cage, only half the voltage used for corresponding tests on the open line.

III. RESULTS OF CORONA LOSS MEASUREMENTS

(1) *General Remarks.* The curves showing the losses as functions of the voltage are characterized by a bend, below which the losses are very small and vary only slightly, and above which they rapidly increase with the voltage.

For dry weather conditions the exact measurement of the voltage corresponding to the bend in the characteristic curve is very difficult. On the other hand, the exact definition of this voltage is essential as it represents, in fact, the upper limit of the operating voltage of the line. If, however, curves are plotted the abscissae of which are not the voltage but the corresponding mean potential gradients (corrected to take account of meteorological conditions), it will be seen that the gradients corresponding to the bends of the various curves, for different mountings of the conductors, deviate by not more than 5 per cent from the theoretical disruptive gradient, i.e. 21 kV/cm.

This is important as a first basis for comparison between the different arrangements, and also as a first datum for the choice of an operating voltage for any given arrangement.

(2) *The Ageing of Conductors.* It has often been noted that the losses on a line which has been in service for a prolonged period are smaller than those on a new line. This phenomenon was measured on phase III of the experimental line. Fig. 8 shows some of the

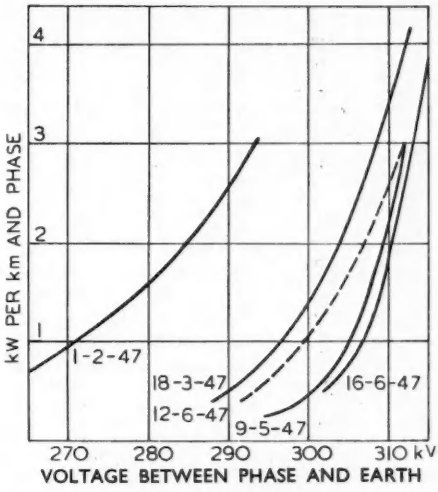


Fig. 8. **Effect of ageing.**
Twin conductor with $D = 40$ cm, horizontal. Note increase of the critical voltage with time.

curves resulting from measurements in dry weather at various dates. It shows that ageing has two effects, namely, to raise the critical voltage (i.e. the bend in the curves) and to increase the slope of the curves which appears to reach a limit of 0·5 kW/kV. It should be noted, however, that if the operation of the line is interrupted for a longer period, the losses after putting it in service again, correspond to those of a newer line (see curve dated 12.6.47), but revert to their previous value within a few hours. These phenomena were found to be due to a layer formed on the conductor which, in the main, consisted of carbonized organic matter and aluminium salts. Ageing must be taken into account when making comparative measurements on transmission lines.

(3) *Influence of Atmospheric Conditions.*

(a) Density and humidity of the air:—

The change of the losses is inversely proportional, while the corresponding variation of the critical voltage is directly proportional, to the variation of the air density. Variations of the humidity have no effect on the critical voltage or on the losses. If barometric corrections are made, the losses measured as functions of the voltage on various occasions under fair weather conditions, show only very little divergences.

(b) Bad weather:—

Curves showing corona losses under bad weather conditions (rain, fog, snow, sleet) as functions of the voltage, can be assumed to be composed of two superimposed components, one representing an exponential and the other a parabolic function. (Fig. 9).

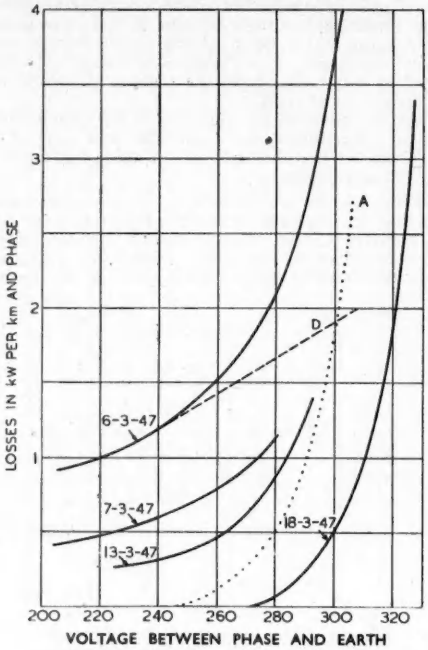


Fig. 9. Effect of atmospheric conditions.
D = 20 cm, horizontal.
18/3/47: fair weather.
13/3/47: wet.
7/3/47: 4 hours after snowfall.
6/3/47: immediately after rainfall.
This curve may be divided into components A for discharge losses and D for dielectric losses.

(i) The first component is not affected by the weather and follows an exponential function, at least up to a point 20 to 30 kV above the critical voltage. This component is influenced by the ageing of the line; it is also influenced by atmospheric disturbances although the slope of the steep part of the curve is not influenced by this. In the following it is referred to as the "exponential component."

(ii) The second component exists in practice only under bad weather conditions, as a function of the voltage, and is represented by a parabola. This component may, in exceptional circumstances, also occur during fair weather, in which case it is explained by the presence of smoke or dirt invisible to the naked eye. This component is referred to as the "parabolic component."

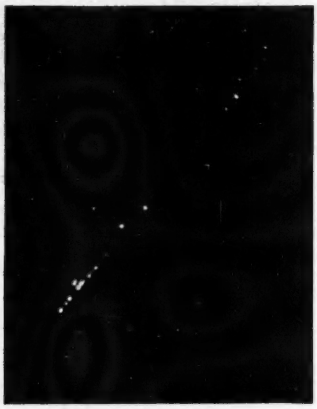


Fig. 11. Luminous discharge on phase I.
U = 310 kV, D = 60 cm vertical, in rain, at period of maximum losses.

Fig. 12. Same as Fig. 11, towards end of rain.

The measurements were carried out on one and two phases at a voltage of 250 kV between phase and earth, which corresponds to about 400 kV in 3-phase operation. This voltage was kept constant as measurements were taken under varying weather conditions.

In rain the losses seem to depend not only on the quantity of the precipitation but also on the size of the rain drops and the degree of cable wetting. Rain generally produces a very rapid increase of losses reaching a peak which lasts only a few minutes, after which the losses decrease and are very unstable throughout the remaining duration of the rain. When the rain ceases the losses still persist for a certain time, corresponding to the drying period of the cable, which may vary from a few minutes to half an hour. Fig. 10 represents a registration showing, at the commencement, two short but intensive rain bursts, followed by persistent but variable rainfall with subsequent mist. Figs. 11, 12 and 13 show corresponding luminous corona discharges, the photographs having been taken,

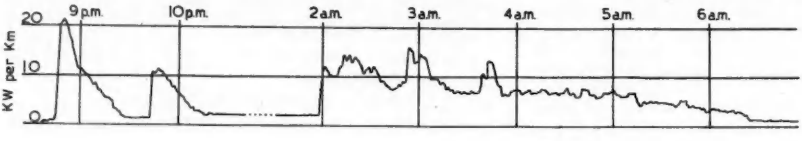


Fig. 10. Losses on phase III.
D = 40 cm horizontal; U = 255 kV, phase/earth; two-phase operation (I-III); rain at nightfall, morning fog.

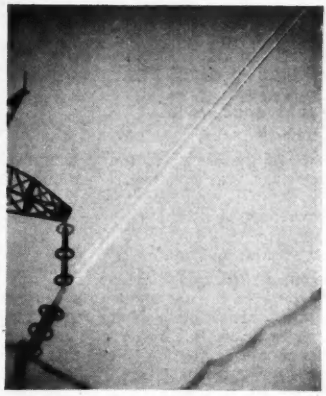


Fig. 13. **Same as Fig. 11, during drying.**

however, under similar conditions on a different occasion.

The existence of these three distinct periods may be explained as follows:—

At first the cable is dry and the raindrops falling on the cable adhere to it, while at the same time becoming pointed (through electrostatic forces) and effective as discharge points. This explains the large number of streamers appearing on the conductors and the high losses.

When the cable is completely wet, only the water drops hanging from the cable form discharge points. These drops repulse each other through electrostatic forces and therefore space themselves at equal distances, as was shown by experiments in the cage. The distances between the drops depend not only on the type of cable but also on the intensity of the rain, the wind velocity, etc., which explains the great instability of losses during this phase. During the drying phase the decrease of the number of droplets on the conductors may be assumed to follow an exponential law, as does the decrease of the corona losses.

The losses appear to be higher the smaller the droplets are, and the highest losses are caused by fog.

(4) *The Influence of the Disposition of the Two Conductors.* Comparison of the influence of the disposition of the two conductors, i.e., of the spacing as well as of the vertical or horizontal arrangement, was made by taking measurements in which one outer phase was kept unchanged for reference purposes, while changes were made on the other outer phase, and the central phase remained earthed. The difference in "ageing" of the reference and test phase, respectively, was taken into account.

(a) *Influence of the Distance between the Two Conductors.* In rain, the losses on the test line for spacings of 20, 30 and 40 cm are very close to the losses measured on the reference line with a 40 cm spacing.

In fair weather, the losses with a spacing of 20 cm are slightly smaller than those measured with a spacing of 30 cm which, in turn, are slightly smaller than those for 40 cm.

With a spacing of 60 cm, on the other hand, the losses in fair weather, and more so in rain, are distinctly higher than those for 40 cm, and the difference is even more pronounced for a spacing of 80 cm.

The general conclusion reached from the measurements taken is that losses increase slightly with the spacing between the conductors increasing from 20 to 40 cm, after which they increase more and more rapidly with the spacing, until at 80 cm they are a multiple of the losses at 40 cm. At the same time the critical voltage for fair weather decreases more rapidly (Fig. 14). If

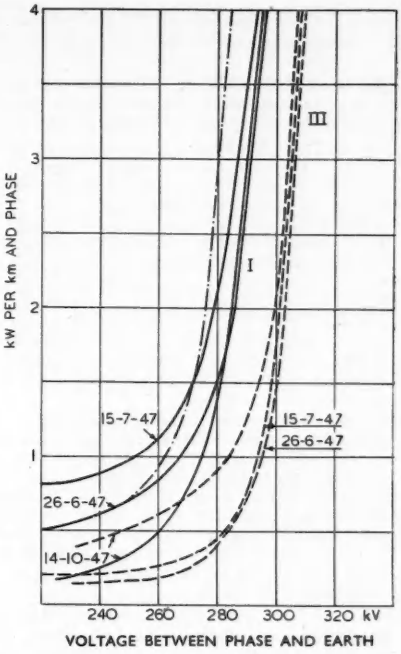


Fig. 14. **Losses as functions of the voltage (single-phase operation)** Phase I: D = 80 cm vertical; phase III: 40 cm horizontal; fair weather.

it is desired to keep a sufficient safety margin for normal operation at 400 kV, a spacing of 60 cm between the two conductors appears to be the upper limit and it is thought advisable to keep well below this figure if excessive losses in bad weather are to be avoided.

It has been shown in section III that the maximum surface gradient of a group of two conductors has a minimum value for a spacing of 29 cm (Fig. 7), while both the minimum and the mean gradients increase with the spacing of the conductors. On the other hand, measurements of losses show a minimum for a spacing of about 20 cm. The behaviour of losses, therefore, follows a function intermediate between the functions determining the maximum gradient and the mean gradient, respectively. It was found that it is possible to describe this function if an "equivalent" gradient is introduced which is the arithmetic mean value of the maximum gradient and the mean gradient.

$$E_q = \tfrac{1}{2}\,(E_m + E_{\max}).$$

This gradient E_q accounts at the same time for variations of the surface gradient round the periphery of the conductors and for the unknown law of the variation of local losses as a function of the local gradients. If it is assumed that this law follows an exponential function it can be shown that the corresponding gradient is

$$E_g = E_m\left(1 + K\frac{2r}{D}\right) = E_q + E_m\,(K - 0{\cdot}5)\,\frac{2r}{D}$$

where K varies between 0·4 (for D = 80 cm) and 0·55 (for D = 20 cm). This formula shows that the calculated values for E_q are always very near the equivalent gradient E_g. It has been confirmed by experiments that the minimum losses do occur at a spacing corresponding to the minimum value of the "equivalent" gradient.

(b) *Horizontal or Vertical Arrangement.* No Difference could be found between the losses for these two

arrangements for spacings between 20 and 40 cm so that both arrangements may be assumed to be equally good.

(c) *Effect of the Ground Wire.* Measurements indicate that, for equal losses, the gradients deviate 1 to 2 per cent from their corresponding theoretical values.

5. *Losses at High Gradients.* At voltages of over 320 kV to earth the rate of increase of losses as a function of the voltage is decreasing, while the losses in rain approach those in fair weather.

At a certain voltage the capacity factor becomes unstable and begins to increase; this can be explained by the occurrence of space charges.

IV. MECHANICAL ACTIONS BETWEEN THE CONDUCTORS

(1) *General.* In the foregoing, the distance between the two conductors had been assumed constant, but this is not generally the case as the distance varies for three reasons:—

(i) The electrostatic repulsion of the two conductors which at every instant are at the same potential and carry equal charges of the same polarity.

(ii) The electrodynamic attraction between the two conductors which carry currents of the same direction and intensity. This force may become very considerable in the case of short circuits when, in addition, the electrostatic repulsion is practically negligible.

(iii) The influence of the wind.

Under certain circumstances the two conductors may not only come in contact with each other, but may remain so for a considerable time. Also, bruises sustained on such occasions are liable to increase the corona incidence, as well as the corona losses for a long period. Considering a transmission line of considerable length, bruises may be numerous, and therefore the total corona losses due to them may attain high values.

(2) *Theoretical Considerations (for Horizontal Disposition).*

(a) *Electrodynamic Forces.* Detailed calculation shows that with increasing current intensity the two conductors approach each other and that definite positions result from individual current intensities. These stable positions are characterized by the corresponding minimum distances between the conductors at the centre of the span. However, from a certain critical intensity I_C onward, no stable equilibrium can be maintained and the conductors approach each other until contact is established. If subsequently the current is reduced, the two conductors still remain in contact until another critical value I_D is reached at which the attraction between the conductors is equal to the force of gravity tending to separate the conductors.

The following table shows the critical currents for the experimental line at Chevilly, and for a sag of 14 meters as calculated by 2 different methods:

Spacing D (cm)	I_C (Amp)		I_D (Amp)	
20	373	406	308	368
30	559	608	435	512
40	746	812	540	644
50	932	1014	640	760
60	1118	1216	738	882

The effect of the sag on the values for I_C and I_D was found to be $\pm$ 3·3 per cent if the sag varies $\pm$ 1 metre.

If n equidistant spacers are used to keep the two wires at a given distance, the figures for the corresponding critical intensities can be obtained by multiplying the figures of the above table by a factor $(n + 1)$.

(b) *Electrostatic Forces.* Taking into account the electrostatic forces, the gravity, the deformation of the cables, and the charges of capacity due to this deformation, the following figures were calculated for a line having a sag of 15 m:—

Distance between the conductors without the application of a potential	20	30	40	50	60 cm
Maximum distance between the conductors on application of 450 kV	38·5	45	52	59·5	68 cm

The presence of spacers restricts the lateral movement of the conductors, and a voltage V produces the same maximum deviation as a voltage $\frac{V}{1+n}$ would do on a similar line without spacers.

(c) *Combined Effects of Electrostatic and Electrodynamic Forces.* It can be shown that under normal load conditions, attraction and repulsion cancel each other out, while below full load repulsion prevails. In the case of a short circuit the two conductors may make contact, and care must be taken to design the line so that the forces of repulsion are sufficient to separate them in the short interval between the clearing of the short circuit and the restoration of the normal load.

(3) *Results of Experiments.*

(a) *Action of the Wind*: for a span of 500 m and the direction of wind perpendicular to the line, a wind speed of 15 m/sec causes two conductors of 40 cm spacing to make contact. The corresponding wind speed for 60 cm spacing is 25 m/sec. With one spacer in the centre of the 500 m span, a wind of 25 m/sec is still sufficient to bring about contact between conductors spaced 40 cm apart, but not between conductors spaced 60 cm apart.

(b) *Electromagnetic Attraction*: the measurements confirm the calculated figures, in particular for a spacing of 40 cm.

(4) *Provisional Conclusions.*

In the case of the horizontal arrangement of the two conductors, contact between them can be brought about through strong wind or through major short circuits. Under normal operating conditions, only the effect of the wind need be taken into account. It appears that contact between the conductors can be entirely avoided if two spacers are installed when the distance between the conductors is 40 cm, and one spacer if the distance is 60 cm. To prevent contact in the case of a short circuit, one spacer is required if the distance between the conductors is 40 cm, and two spacers if the distance is 60 cm.

V. RADIO INTERFERENCE

Measurements have shown that radio interference caused by a three-phase 400 kV line is greater than that from an ordinary 220 kV line, if measured directly below the transmission line. At a lateral distance of about 30 m from this point, radio interference is similar for both types of line; but at a distance of about 50 m, the interference is negligible.

The influence of voltage on radio interference is less than the corresponding influence on corona formation.

VI. ELECTRICAL CHARACTERISTICS OF TRANSMISSION LINES FOR VERY HIGH VOLTAGES

The use of group conductors enables long transmission lines to be operated at a higher load due to reduction of their self-inductance and a simultaneous increase of their capacitance, which results in a net increase of the natural load of the line, if compared with a line strung with hollow conductors of the same useful cross-section. The only disadvantage is the increased reactive power to be absorbed in the case of no-load operation.

Vibrations of Vertical Vane-Type Stirrers

By R. CANTONI. (From *L'Ingegnere*, No. 3, March, 1948, pp. 187-188, 1 illustration.)

THE following investigation applies to an electric motor driving a vertical vane-type stirrer mounted on two parallel, flexible, horizontal beams. The motor mass is capable of performing oscillations in a vertical plane perpendicular to the axes of these supports, but cannot vibrate in a vertical plane parallel to these axes. The central portions of the supporting beams are alternately subjected to periodic variations of height, corresponding to deflections which are equal and opposite in direction at any given instant.

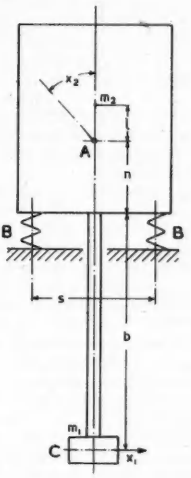


Fig. 1.

Mixers of this type working at speeds above the critical are being manufactured overseas, and the purpose of this article is to develop a method enabling dangerous criticals to be avoided, using the theory of vibrations with two degrees of freedom. For simplicity the effect of the additional accelerating forces acting on the mass of the stirring vanes will be neglected, since this would lead to a system with three degrees of freedom.

The equivalent system (see Fig. 1) consists of an electric motor which can rotate about its horizontal axis A, perpendicular to the paper plane; the motor is mounted on two supporting beams B, and its shaft carries the vane-type rotor C which serves as a stirrer. The system can vibrate with two degrees of freedom in the plane of the figure provided we neglect, as already stated, the additional accelerations which tend to move the stirring vanes out of this plane. Neglecting the mass of the shaft, the following notation will be used:—

m_1 = mass of the stirrer vanes [gr]; m_2 = mass of motor; $\mathcal{J}_2$ = moment of inertia of motor relative to a horizontal axis through its centre of gravity [gr cm²]; n = distance of node from free shaft portion beyond the bearing (see Fig. 1); l = distance between centre of gravity and node [cm]; x_1 = linear deflection of rotor towards the right [cm]; x_2 = angular displacement of motor axis in counter-clockwise direction (radians); p = force on each individual support [dynes]; h = deflection of each of the supports [cm]; d = shaft diameter [cm]; s = distance between the axes of the supports [cm] represented by the springs B in Fig. 1; $\mathcal{J}$ = axial moment of inertia of shaft cross-section [cm⁴]; b = free length of vertical shaft.

We then have

$$-m_1 \frac{d^2 x_1}{dt^2} - k_1 x_1 + k_3 x_2 = 0 \quad .. \quad .. \quad (1)$$

where $k_1 = 3E\mathcal{J}/b^3 = 6 \times 10^{12}\, \pi\, d^4/(64b^3)$ [dynes/cm]; $k_3 = (b + n)\, k_1$ [dynes]; $E_{\text{steel}} = 2 \times 10^{12}$ [dynes/cm²]; furthermore:

$$-(\mathcal{J}_2 + m_2 l^2) \frac{d^2 x_2}{dt^2} + R^3 x_1 - k_2 x_2 = 0 \quad .. \quad (2)$$

where $k_2 = \frac{s^2}{2} \frac{dp}{dh} + (b + n)^2\, k_1$ [dynes cm]

For example, let $b = 60$ cm, $d = 2$ cm, and assume that each support deflects by 1 cm under a load of 204 kg, then the stiffness of the supports is:

$$dp/dh = 2 \times 10^6 \text{ dynes/cm},$$

so that $k_1 = 6 \times 10^{12}\, \pi \times 2^4/(64 \times 60^3) = 2{\cdot}18 \times 10^7$ dynes/cm; it will furthermore be assumed that $s = 20$ cm, $l = 10$ cm, and $n = 0$.

The motor can be represented by an equivalent spherical hollow mass of cast-iron (density 7·5) with an outer radius of 10 cm and a wall thickness of 4 cm. Then

$$\mathcal{J}_2 = 7{\cdot}5 \times 8\, \pi\, (10^5 - 6^5)/15 = 1{\cdot}162 \times 10^6 \text{ [gr cm}^2\text{]}$$
$$m_2 = 4\pi \times 7{\cdot}5\, (10^3 - 6^3)/3 = 2{\cdot}47 \times 10^4 \text{ [gr]}$$

In addition, we have

$$k_3 = b_1\, k_1 = 1{\cdot}305 \times 10^9 \text{ [dynes]}$$

and $k_2 = s^2\, dp/(2dh) + b^2 k_1 = 1{\cdot}1182 \times 10^{11}$ [dynes cm]

Putting $x_{1,2} = a_{1,2} \sin \omega t$, we obtain from eqs. (1) and (2) the characteristic equations:

$$\begin{cases} m_1 a_1 \omega^2 - k_1 a_1 + k_3 a_2 = 0 \\ k_3 a_1 - k_2 a_2 + (\mathcal{J}_2 + m_2 l^2)\, \omega^2 a_2 = 0. \end{cases}$$

The frequency equation is obtained from the determinant of the coefficients of a_1 and a_2:

$$m_1 (\mathcal{J}_2 + m_2 l^2)\, \omega^4 - [k_1 (\mathcal{J}_2 + m_2 l^2) + k_2 m_1]\, \omega^2 + k_1 k_2 - k_3^2 = 0.$$

This gives the solution

$$\omega^2 = \frac{1}{2m_1 (\mathcal{J}_2 + m_2 l^2)} \left\{ k_1 (\mathcal{J}_2 + m_2 l^2) + k_2 m_1 \pm \sqrt{[k_1 (\mathcal{J}_2 + m_2 l^2) + k_2 m_1]^2 + 4m_1 (\mathcal{J}_2 + m_2 l^2)(k_3^2 - k_1 k_2)} \right\}$$

It will be observed that both these values of ω^2 tend towards the limiting value k_1/m_1 which would be obtained with a shaft fixed at one end, in the first case when the moment of inertia of the motor tends towards infinity, and in the second when the rigidity of the supports becomes infinite.

The calculation with the numerical values already used gives

$\omega^2 = 29{,}800 \pm 24{,}300$ rad/sec, and these values correspond to the two frequencies

$$N_1 = 2{,}240 \text{ rpm}$$
$$N_2 = 709 \text{ rpm}$$

If the node of the vibration is displaced to the centre of the motor, so that $l = 0$, and $n = 10$ cm, we obtain $k_2 = (b + n)\, k_1 = 1{\cdot}525 \times 10^6$ [dynes]

$k_2 = s^2\, dp/(2dh) + (b + n)\, {}^2k_1 = 1{\cdot}47 \times 10^{11}$ [dynes cm]

and these values correspond to the critical frequencies

$\omega_1 = 386$ rad/sec or $N_1 = 3{,}670$ rpm

$\omega_2 = 77$ rad/sec or $N_2 = 735$ rpm

This shows that a vertical displacement of 10 cm of the node of the vibrations has left one of the two resonant frequencies nearly unchanged but has nearly doubled the higher frequency and removed it considerably from the 2,400 rpm of the two-pole 42 cps motor, and there is no doubt that its efficiency is considerably improved as a result.

With absolutely rigid supports, the resonant frequency which is determined by $\sqrt{k_1/m_1}$ would be 1580 rpm.

Most manufacturers are consequently endeavouring to bring the centre of gravity of their stirrer motors as far down as possible, both in the case of the motor mounted on flexible supports, as in the above example, and in the case of the motor fixed on to the flat cover of a container.

In certain cases it has been found convenient to mount slightly below the main stirrer vanes an auxiliary stirrer attached to it and having the appearance of a small pulley with a vertical axis.

New Type of Electrode Coating for Welding Low-Carbon Steel

By A. A. Alov. (From *Avtogennoye Delo, Russia*, No. 9, 1947, pp. 8-11, 1 illustration.)

Electrode coatings can be sub-divided into three main groups:—

(1) coatings consisting mainly of iron oxides.
(2) coatings containing a considerable proportion of organic substances.
(3) coatings containing carbonates, silicates and ferro-alloys.

The first group is the one most extensively used and some very good general-purpose electrodes of that type are used on a large scale.

For production purposes, however, general-purpose electrodes are of no great importance, as horizontal welding is the most usual procedure.

The object of research described in the report was to develop a suitable electrode coating, particularly for the horizontal welding of steel parts, with a view to obtaining a slag composition of FeO-MnO-SiO_2.

Red iron ore (hematite) was used, having the following composition:—

Table I

Chemical Composition of Hematite in per cent.

Fe_2O_3	SiO_2	Al_2O_2	MnO	MgO	CaO	SO_3	P_2O_5
95·03	2·34	1·71	0·06	0·23	trace	0·38	0·03

To obtain the necessary manganese oxide, manganese ore could be used, but it was found that the latter dissociates at temperatures as low as 470° C, giving up free oxygen which oxidizes the organic components of the coating; it was decided, therefore, to use ferromanganese instead. Ferromanganese was found to be advantageous because it generates a large amount of heat when oxidized, i.e. 10,400 kcal for 100 grams. Further tests showed that ferromanganese of low carbon content was superior to blast furnace ferromanganese, and the former was, therefore, adopted, its composition being as follows:—

Table II.

Chemical Composition of Ferromanganese in per cent.

C	Mn	Si	S	P
1·39	78·25	1·50	0·010	0·34

In order to obtain the necessary concentration of silica, glass sand was added. Feldspar and marble were also added, in order to assist in obtaining a stable welding arc. The compositions of all the above constituents and of soluble glass are stated in the report.

The optimum proportions of the listed constituents of the coating were investigated experimentally and were found to be as follows (in per cent by weight):—

Hematite	33
Sand	24·6
Marble	4·4
Feldspar	6·3
Ferromanganese	27
Starch	4·7
Soluble Glass (as a percentage of the total of other constituents)	25-30

The coating was applied by immersion or by pressing it on to the electrode wire.

The recommended thickness of coating is given in Table III.

Table III.

Recommended Thickness of Coating.

Diameter of wire, mm	4	5	6
Thickness of coating, mm	1·1-1·3	1·3-1·5	1·5-1·7

After coating, the electrodes were dried for 12 to 24 hours (depending on temperature and moisture content of the air) and then heated in a furnace at 180-220° C for 1½ hours.

The most suitable composition of each of the constituents yet remains to be determined more accurately, but the following compositions are recommended pending more detailed investigation:

Hematite (in per cent): $Fe \geqslant 66$; $SiO_2 \leqslant 2{\cdot}5$; $SO_3 \leqslant 0{\cdot}40$; $P_2O_5 \geqslant 0{\cdot}1$. It is essential to grind the hematite to pass through a 1600/cm² mesh.

Sand (in per cent): $SiO_2 \geqslant 96$; $S \leqslant 0{\cdot}05$; $P \leqslant 0{\cdot}05$.

The sand should be ground to a size below 0·1 mm dia.

Marble (in per cent): $CaO \geqslant 50$; $SiO_2 \leqslant 1{\cdot}0$; $S \leqslant 0{\cdot}1$; $P \leqslant 0{\cdot}05$.

The marble is ground to a size equal to that of the hematite.

Feldspar (in per cent): $SiO_2 = 64$-68; $Al_2O_3 = 18$-21; $Na_2O + K_2O = 10$-15; $S \leqslant 0{\cdot}01$; $P \leqslant 0{\cdot}05$; ground to pass through a 1600/cm² mesh.

Ferromanganese (in per cent): $C \leqslant 2$; $Mn \geqslant 75$; $SiO_2 \leqslant 2{\cdot}5$; $S \leqslant 0{\cdot}025$; $P \leqslant 0{\cdot}4$; ground to pass through a 1600/cm² mesh. It is undesirable to have finer particles of ferromanganese, as this reduces the quality of the coating.

Soluble Glass (in per cent): $SiO_2 = 32$-$34{\cdot}5$; $Na_2O = 11$-$13{\cdot}5$; water ≈ 57; $S \leqslant 0{\cdot}07$; weight ratio of SiO_2 to $Na_2O = 2{\cdot}6$-$3{\cdot}0$.

The new electrodes have been subjected to very extensive tests, and the composition and strength of welds and the composition of slag were also examined. Detailed test results are stated in the report and indicate the high quality of the electrodes. On a large number of test pieces not a single crack in the seams or craters was detected. In comparative tests using other Russian and imported electrodes, under the same conditions, cracks were detected in several cases.

The new electrodes have a further valuable quality in that the slag can be easily removed from the surface of the seam; it is found to be brittle and to come off when the seam has cooled down.

Method of Pumping by means of Compressed Air based on Hydrostatic Pressure

By J. P. POLOUEKTOFF. (From the 40th Annual Session of *Association Maritime et Aeronautique*, May 28, 1948, 24 pages, 6 illustrations.)

THE performance of pumping operations in salvage work, particularly at great depth, has always presented considerable difficulties. The last World War has rendered the problem of raising wrecks particularly important, and in French ports alone at the end of hostilities there remained more than 2 million tons of sunken vessels. It, therefore, became necessary to develop pumping techniques on a large scale, utilizing the most expeditious means. A search for a robust, simple and reliable type of submersible pump, however, proved unsuccessful, as the only available type, which was the electrically driven underwater type, is not suitable either for operation in open water or even in roadsteads. It is for this reason that the author decided to attempt the design of a pump that could be operated by compressed air. In a pump of this type, suction is replaced by the hydrostatic pressure due to immersion, while delivery is provided by the pressure of the compressed air, the discharge action resembling the displacement process effected in a sluice or a diver's bell. Methods of this nature are well known, but have not hitherto been employed for continuous pumping processes. The adoption of this principle makes it possible to construct a pump of small dimensions, using air of low pressure. The use of low-pressure air is a factor of great importance, as it permits the attainment of an average efficiency of 25 per cent in the case of an installation where there is a free escape of air, and a theoretical efficiency of 44 per cent where provision is made for air recovery.

For the prototype commissioned by the Directorate of Naval Construction the following specifications were laid down: Overall diameter 0·72 metre, overall height 1·60 m, maximum delivery 200 m³/hr, the pump being immersed flush with sea level; delivery height 25 m, depth of immersion 25 m, and maximum compressor power 80 hp.

The salient design features of the pump are shown in Fig. 1. Here C_1 and C_2 are two concentric cylinders, the working volume of the compensating cylinder C_2 being about 40 per cent larger than that of the main cylinder C_1. Working air distribution is effected by the piston valve D. The water is admitted to the cylinders through the suction valves A_1 and A_2, respectively, and is discharged through the delivery valves R_1 and R_2 respectively. Operation of the air distributing valve D is effected by the pilot cylinder E, the piston valve of which is activated by the double float F. A snap-action mechanism is incorporated in the link gear connecting float F with the pilot valve, which forces the latter from one extreme position to the other in accordance with the position of the float, thus precluding any intermediate valve positions. There is also a hydrostat K which throttles the air admission pressure to the required value.

Assuming the air distribution valve D to be in its topmost position, compressed air will be admitted from the air supply line into the inner cylinder C_1, from whence the water will be expelled by the air pressure through the discharge valve R_1; the water admission valve A_1 will remain closed. The fall in the water level in the cylinder C_1 will be accompanied by a descent of the float F. While the water is being discharged from tank C_1, water will flow into the outer cylinder C_2 through the inlet valve A_2, the compressed air being allowed to escape from this cylinder. When the water in C_1 has fallen to its lowest level, the float F will shift the pilot valve by snap-action into its appropriate extreme position, which, in turn, will cause the distribution valve D to descend to its bottom position. Air

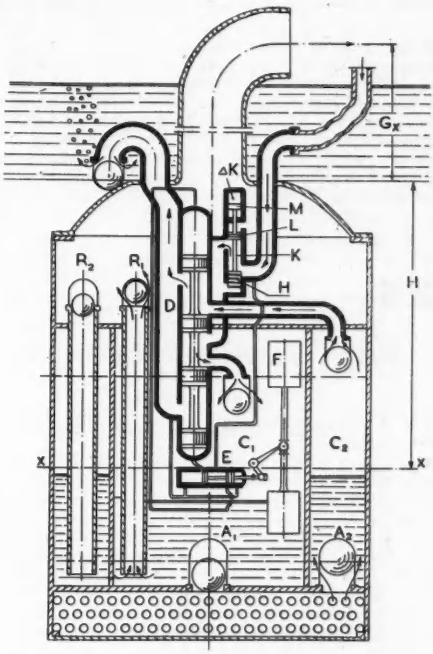


Fig. 1

will now be admitted into the outer cylinder C_2 and at the same time the compressed air will be allowed to escape from cylinder C_1, causing the admission valve A_1 to open. Thus, cylinder C_1 will be filled, while cylinder C_2 is being emptied, and so on.

The action of the hydrostat K consists in automatically controlling the air pressure P' in accordance with the discharge head G_x, the depth of immersion H and the pressure drop $\Sigma\Delta r$ in the discharge pipe. The hydrostat comprises a piston valve L which is subjected on one side, to the air pressure P prevailing in the pump interior and, on the other, to the hydrostatic pressure $(G_x + \Sigma\Delta r)$. The counterweight Δk corresponds to the pressure drop $\Sigma\Delta k$ occurring in the discharge system within the pump. This device ensures constant delivery at all depths by admitting to the cylinders the amount of air required to ensure the displacement of the water at constant speed. The counterweight Δk, therefore, governs the time t_r required for a single discharge from the cylinders to take place. Smooth operation of the hydrostat valve is ensured by the provision of the dashpot M. The author gives a complete analysis of the operating cycle of the pump including numerical values for a pump for the aforementioned delivery of 200 tons per hour. Tests carried out in April and May, 1947, are stated to have fully confirmed the theoretical calculations. Typical test results are given in the Table below, where H is the depth of immersion based on the datum level x — x in the pump, H_d the dynamic pressure required to ensure the discharge of the water through the delivery lines and valves at a given velocity, $\Sigma\Delta k$

H m	G_0 m	G_z m	H_d m	$\Sigma\Delta k$ m	$\Sigma\Delta r$ m	$\Sigma\Delta p$ m	P kg/cm²	E_T hp	E_A hp	η %	Delivery m³/hr
10	1·00	11	0·975	2·019	1·060	1·200	1·625	8·6	30	28·5	210
14	1·00	15	0·975	2·019	1·070	1·302	2·046	11·5	45	25·6	210
18	1·00	19	0·975	2·019	1·350	1·410	2·475	14·7	55	26·8	208
20	1·00	21	0·975	2·019	1·520	1·520	2·700	16·0	65	24·8	206
22	1·00	24	0·975	2·019	1·650	1·632	2·827	18·6	68	27·4	208

the sum total of the pressure drops in the internal discharge system, $\Sigma\Delta r$ the sum total of pressure drops in the discharge system external to the pump, G_0 the elevation of the top of the pump above the datum level $x - x$ and G_z the static head above the top of the pump as indicated in the accompanying drawing. The sum total of the loss in air pressure between compressor and pump is given by $\Sigma\Delta p$. Thus, the total air pressure P required at the compressor outlet is given by $P = Hd + \Sigma\Delta k + \Sigma\Delta r + G_0 + G_z + \Sigma dp$ and is shown in the table in units of kg/cm². The theoretical power consumption is represented by E_T and the actual power consumption by E_A, while η is the pump efficiency in per cent. The prototype pump was used at Toulon on the occasion of the raising of the sunken mail steamer "Kairouan." After 13 months' operation in both salt water and fresh water, the pump was opened for inspection and found to be in perfect condition.

RUSSIA

New Universal Inductor for High Frequency Hardening of Steel

By V. V. ALEXANDROV and S. M. GAMAZKOV. (From *Vestnik Machinostroenia*, No. 2, 1948, pp. 43-46, 2 illustrations.)

INDUCTORS for high-frequency case-hardening are normally built up copper tubes of round or other cross-sectional shape. The main limitation to the application of these inductors is the fact that they must very closely follow the size and shape of the object to be hardened. The article deals with single-coil inductors where the further difficulty arises of shaping the inductors accurately. Inaccurate shaping causes uneven hardening, particularly when case-hardening or hardening by "self-cooling," as both these methods require high local concentration of energy. Cooling water requirements determine the minimum permissible depth of the inductor and it has been established that about $\frac{5}{16}$-in is the practical limit, thus restricting the heated area.

This investigation is concerned with case-hardening to a depth of the order of 0·01 to 0·04-in and, further, with the "self-cooling" method of hardening. In the latter case, the heat generated in the thin top layer of the metal is dissipated rapidly into the cold metal adjoining it, causing the surface to harden. The universal inductor designed by the authors is shown in Fig. 1. It is capable of hardening a wide range of shapes and sizes and consists of the main structure and interchangeable inserts to suit different shapes and sizes of jobs. The main structure is made of copper tube and has lugs for connection to the H.F. supply, and protruding nozzles for the cooling water supply. A solid split copper ring is welded inside the inductor, acting as a seat for its interchangeable insert. The insert is a flat slotted ring and acts as the working part of the inductor. It is made of copper and is machined to ensure satisfactory contact with the outer ring. The outer ring is dowelled in order to locate the insert accurately. The inserts can be of different shape and depth and permit the use of a single inductor for a range of parts, with diameters ranging from $1\frac{3}{16}$ to $5\frac{1}{8}$-in and weights from $\frac{1}{16}$ to $1\frac{9}{16}$-in.

The same inductor can also be used for gradual case-hardening of large parts by moving the inductor so as to cover the whole of the surface. Comparative tests carried out with the universal and standard type of inductors have given satisfactory results. It was found that no sparking occurred when the insert had a thickness of 0·39-in or more, and no additional contact pressure by the use of screws was required. The contact surfaces must, however, be protected from oxidation and it is advisable to cover them with a non-oxidizing metal.

The rating of the H.F. generator used for testing was approximately 60 kW, time of application 6 to 10 seconds, the case-hardening depth being 0·06 to 0·12-in and Rockwell hardness 58-66.

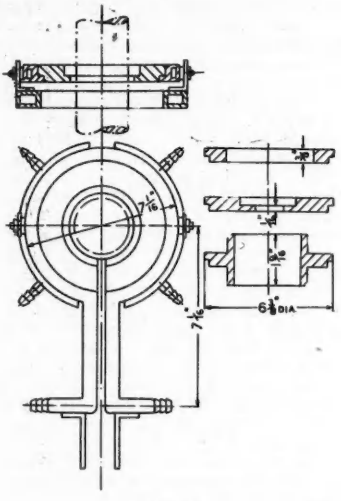


Fig. 1

When case-hardening a large part, the depth of hardening ranged from 0·02 to 0·06-in, depending on the speed at which the inductor was moved. The depth of the hardened layer also depends on the frequency of the generator, a higher frequency producing a thinner hardened layer, all other conditions remaining equal.

The tests proved further that a high degree of concentration of energy is possible—a condition particularly essential for hardening by "self-cooling."

The Ray Transformer

By R. WINDERÖE (From *Schweizer Archiv*, Vol. 13, No. 8, August, 1947, pp. 225-232, and No. 10, October, 1947 pp. 299-311: 14 illustrations.)

Continued from July Issue.

For radial oscillations, the differential equation of the electron motion is as follows:—

$$m\ddot{r} + \dot{m}\dot{r} + mv^2 r(1-K)/R_0^2 = 0 \quad .. \quad (34)$$

or $$\ddot{r} + \dot{U}\dot{r}/(U+\epsilon) + \omega^2(1-K)r = 0 \quad .. \quad (34a)$$

where the dot signifies d/dt and ω is the angular velocity of the fundamental motion of the electrons. Consequently, the period of the electron oscillations is $2\pi/\sqrt{1-K}$ (i.e. when $K = {}^1/_2$, the period is 508 deg., and the half period 254 deg.). The action integral of the oscillations is

$$\oint m\dot{r}\,dr = \pi r_m^2 mv\sqrt{1-K}/R_0 \quad .. \quad (35)$$

As the action integral must remain constant when the voltage increases, the adiabatic decrease of the oscillatory amplitude with increasing voltage will be given by

$$dr_m/r_m = -dv/2v = -dB/(2B) = -dU/(4U) \quad (36)$$

It is only because the disturbing oscillations are thus damped out that the electrons are able to perform so many revolutions. Damping is therefore an important property of the non-material circular conductor.

The theory of the circular accelerating tube thus far developed is due to Steenbeck, Kerst, and Serber, and as a first approximation of the problem it only holds, in fact, for the immediate vicinity of the equilibrium orbit. Owing to the limited width of the guide-poles, the cross-section of the circular conductor will also be limited; the radial stabilizing forces will therefore reach a maximum value and then decrease to zero at greater distances. For the stabilizing forces we could therefore assume, for instance, a relation of the type

$$P_{stab} = |\ a_1 r - a_2 r^n\ | \quad .. \quad .. \quad (37)$$

In the following a simple parabolic form will be used for the curve of the stabilization forces:

$$P_{stab} = \left| P_{stab\,m}\left(2 - \frac{r}{r_M}\right)\frac{r}{r_M}\right| \quad .. \quad (37a)$$

It can easily be shown that this corresponds to a potential curve which is a cubic parabola when $r > 0$:

$$eV_{stab} = \int_0^r P_{stab}\,dr$$

$$= P_{stab}\,r_m^2\left[\left(\frac{r}{r_M}\right)^2 - \frac{1}{3}\left(\frac{r}{r_M}\right)^3\right] \quad (38)$$

The so-called stabilization potential can generally be derived from the Hamiltonian expression for the energy of the disturbed electrons. Expanding eq. (18b) we obtain (with p_r = momentum of the radial oscillations):

$$H = eU = c\sqrt{m_0^2 c^2 + p_r^2 + \left(\frac{p_\varphi}{R} + eA_\varphi\right)^2} - m_0 c^2$$

$$\sim \frac{p_r^2}{2m} + \frac{e^2}{2m}\left(\frac{p_{\varphi_0}}{eR} + A_\varphi\right)^2 \text{ (N.R. region)} \quad (39)$$

The term $p_r^2/(2m)$ corresponds to the oscillatory kinetic energy of the electrons, while the second term represents a "potential" energy of the electrons. For the stabilizing potential field we therefore obtain the general expression

$$V_{stab} = \frac{e}{2m}\left(\frac{p_{\varphi_0}}{eR} + A_\varphi\right)^2 \text{ (N.R. region)} \quad (40)$$

which in the case of correct-phase electrons ($p_{\varphi_0} = 0$) coincides with the expression (38a) apart from the integration constant $V_{stab}\,R_0$. Fig. 5 shows a (slightly exaggerated) example of the variation of the potential curves with various values of p_{φ_0}. For correct-phase electrons ($p_{\varphi_0} = 0$), the minimum of the potential curve, corresponding to the equilibrium orbit R_0, is situated approximately half-way between the two maxima of the curves, and in this case these maxima determine the external limits of the non-material conductor.

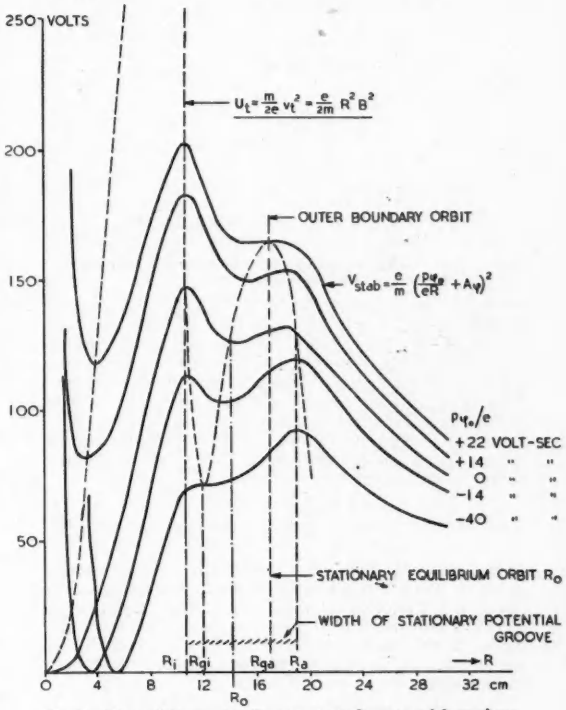


Fig. 5. Potential curves of a ray transformer with various values for the initial momentum P_{φ_0} of the electrons.

When p_{φ_0} increases the equilibrium orbit grows larger, the minimum of the potential curve is displaced outwards, and the depth of the potential groove also decreases. If the potential curve finally acquires a point of inflexion, the potential groove is then completely flattened, and the phase error p_{φ_0} reaches its maximum permissible value; the same applies to the equilibrium orbit, which in this case will coincide with the external circular boundary R_{ga}. In the potential groove of the guiding field the electrons move more or less in the same way as smal spheres rolling forwards in a material groove and which are also capable of performing transverse

oscillations, due to disturbing forces, between the boundary edges of the groove. The space-charge potential of the electrons will be superimposed on the groove potential, and under equilibrium conditions the electrons will be so distributed that the resulting potential curve will be horizontal within the groove right up to the boundary potential.

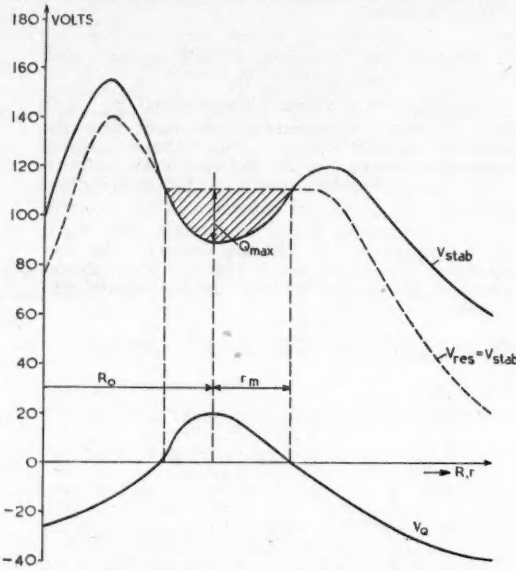


Fig. 6 **Potential groove V_{stab} of a ray transformer with a superimposed space-charge potential V_a.**

Fig. 6 shows an example of a potential groove filled in this manner and one sees that the space-charge potential can be likened, to a certain extent, to a volume of fluid in a material groove. The maximum space charge for a charged cylinder of radius r can therefore be calculated by equating the forces of the space charge to the stabilization forces. If a linear increase of the forces is assumed, according to the Steenbeck-Kerst-Serber theory based on a parabolic groove, the maximum charge is then given by

$$Q_{max} = \frac{8\pi^2 r^2 (1-K)\,\epsilon_0\, U}{R_0}\left(1+\frac{U}{2\epsilon}\right)\left(1+\frac{U}{\epsilon}\right) \quad (41)$$

In the case of the potential groove which is unlimited, on account of r in this first approximation, the maximum charge is not affected by phase error. For the parabolic force curve of the limited potential groove the maximum charge of the correct-phase electrons for a radius r of the space charge (r_M = distance of the maximum stabilizing force):

$$Q_{max}{}^{\star} = Q_{max}\left\{1-(r/2r_M)\right\} \quad .. \quad (42)$$

When $r = 4r_M/3$, the charge $Q^{\star}$ has its highest value, which is one-third of the corresponding "linear" charge Q. For instance, a ray transformer designed for $U_m = 15$ Mv, with $R_0 = 14$ cm, $r = 1$ cm, $r_m = q$ cm, $K = {}^1/_2$ for an initial voltage of 10 kilovolts, can receive a maximum charge $Q^{\star}{}_{max}$ of $1{\cdot}87 \times 10^{-9}$ coulombs in its accelerator tube. At a frequency of 50 cycles/sec this corresponds to a mean current intensity of 0·94 ampere, while the current rotating in the tube when $v \simeq c$ is about 0·64 amperes.

The problem of injecting electrons into the potential groove of the accelerator tube involves considerable difficulties. If the source of electrons is placed within the groove, then there is the possible danger that the electrons will impinge upon this material obstacle during one of their revolutions. If it is attempted to inject the electrons from the outside, over the edge of the potential groove, then after one oscillation in the groove, they will fly out again over the threshold potential.

It is thus soon apparent that with a stationary potential groove there are no possible methods of introducing electrons, and only by varying the groove over an interval of time will it be possible to inject electrons into the accelerator tube. This can be shown simply enough by bringing the source of electrons within the groove (for instance near one edge). The electrons, assumed to be of correct phase, will then perform oscillations about the equilibrium orbit with the amplitude r_1. If they are also capable of oscillating in the axial direction (with the amplitude z_1), then the probability that after n oscillations they will strike the electron source is

$$W = \frac{n}{\pi^2}\left\{\arcsin 1 - \arcsin(1-b)\right\}\left\{\arcsin h - \arcsin(-h)\right\}$$

$$\simeq \frac{2n}{\pi^2}\sqrt{2b}\,\arcsin h, \quad .. \quad .. \quad .. \quad .. \quad (43)$$

where b = width/$(2r_1)$ = relative width, and h = height/$(2z_1)$ = relative height.

Owing to the increase of the stabilizing forces (and of the stabilization potential) the amplitudes of the electron oscillations decrease adiabatically and, after n_1 revolutions,

$$n_1 = 4bU/(\Delta U) \quad .. \quad .. \quad .. \quad (44)$$

where ΔU = increase of the electron potential U per revolution, the electrons will leave the dangerous zone of impingement. The loss through captured electrons per dn revolutions, is after n revolutions:

$$dQ = QW_0\sqrt{\frac{n_1-n}{n_1}}\,dn.$$

If this loss is integrated from $n = 0$ to $n = n_1$, then the efficiency of the electron injection is

$$\eta = Q_1/Q_0 = \exp\left(-\frac{2n_1 W_0}{3}\right) =$$

$$= \exp\left(\frac{-16\sqrt{2}\,U b^{3/2}\arcsin h}{3\pi^2\,\Delta U}\right) \quad (45)$$

This calculation is only valid for the first (Steenbeck-Kerst-Serber) approximation; it shows that the electron injection efficiency decreases when the area of the electrodes is increased, and also that a rapid increase ΔU of the electron potential U has a favourable effect. For instance a ray transformer for a maximum voltage of 15 Mv at $f = 50$ cycles/sec will give the following values: $U = 10$ kV, $U = 17$ V, $b = 0{\cdot}1$, $h = 0{\cdot}4$, $n_1 = 235$ revolutions, $W_0 = 3{\cdot}72$ per cent and $\eta = 0{\cdot}30$ per cent. The injection efficiency decreases very rapidly with increasing values of the supply potential U.

For wrong-phase electrons similar conditions are valid. The electron motion consists, in this case, of an adiabatically damped oscillatory component, and an additional slowly-decreasing exponential term for the equilibrium radius. The resultant electron motion is therefore away from the generating electrodes, and for the injection efficiency we obtain equations similar to those given above (eq. (45)). By applying eq. (45) to wrong-phase electrons (this being permissible for $R \leq R_0 + 0{\cdot}7\,r_{max}$), and assuming the charging time is calculated from eq. (23), we have, with a constant injection current, for the charge of an accelerator tube

which is only very slightly filled: $Q = \text{const.}\ \sqrt{U_a}\ \exp(-U_a/A)$, where $A = 3\pi^2\ \Delta U/(16\sqrt{2}\ b^{3/2}\ \arcsin h)$.

If the electrons are injected from the outside, over the edge of the potential groove, with a suitable radial momentum p_r, then the steadily increasing depth of the potential groove will also make a capture of the electrons possible. Considering first the conditions in the case of an absolutely flat potential groove, which has degenerated into a point of inflexion ($p_{\varphi_0} = p_{\max}$), if we introduce the co-ordinates $\rho = R - R_g$ and $\tau = (t - t_a)/t_a = nB_0\ \Delta U/(B_g\ 2U)$, assuming that the number of electron revolutions $n = 0$ when $\tau = 0$, we can substitute the following approximate value for the stabilization potential

$$V_{\text{stab}} = \frac{e}{2m}\left[\frac{p_{\varphi_0}}{eR} + \left[R_g\,B_g\,(1 - a\rho^3) - \frac{p_{\varphi_0}}{eR}\right](1+\tau)\right]^2 \quad (46)$$

For small intervals of τ the corresponding radial force is then approximately

$$P_{\text{rad}} = edV_{\text{stab}}/d\rho$$

$$= -\frac{e^2}{m}\,R_g\,B_g\left[3a\,R_g\,B_g\,\rho^2 - \frac{p_{\varphi_0}}{e\,R_g^2}\,\tau\right] \quad (47)$$

This stabilizing force follows a parabolic curve, which will now also vary with time owing to the term p_{φ_0}. The differential equation of the electron motion becomes:

$$\ddot{\rho} = \frac{e^2}{m^2}\,R_g\,B_g\left[3a\,R_g\,B_g\,\rho^2 - p_{\varphi_0}\,\tau/(eR_g^2)\right] \quad (48)$$

which can be reduced to the Painlevé form:

$$y'' = y^2 - x \quad .. \quad .. \quad .. \quad (49)$$

(By means of the substitution $y = \rho\,\alpha^{3/5}\beta^{2/5}$, and $x = \tau\,\alpha^{1/5}\,\beta^{1/5}$, we can reduce a differential equation $\ddot{\rho} = \alpha\,\rho^2 - \beta\tau$ to the normal form $y'' = y^2 - x$). Some solutions of this differential equation (according to W. Bothe) corresponding to the various initial conditions of the injected electrons, are shown in Fig. 7. Curves *a*, *b*, *e* and *f* show the paths of the electrons which are captured, while curves *c* and *d* do not lead to capture. The parabola $y^2 = x$ of Fig. 7 corresponds to the extreme positions (outer edge and deepest position) of the potential groove, and this shows that the width, and hence the depth, of the potential groove steadily increases. To determine the variation of the groove it is necessary, however, to have a more accurate approximation than eq. (48), which will also be valid for larger intervals. From eq. (46) we have

$$\ddot{\rho} = \frac{e^2}{m^2}\,R_g\,B_g\,(1+\tau)^2\left[3a\,R_g\,B_g\,\rho^2 - \frac{p_{\varphi_0}}{eR_g^2}\,\frac{\tau}{1+\tau}\right] \quad (50)$$

and the extreme positions of the potential curve can be calculated from

$$\rho_e = \pm\sqrt{\frac{p_{\varphi_0}}{3ae\,R_g^3\,B_g}\,\frac{\tau}{1+\tau}} \quad .. \quad .. \quad (51)$$

The electron motion can now be calculated for other values of p_{φ_0} as well, by means of eq. (50), provided the initial conditions of the electrons are so chosen that we introduce them into an already formed potential groove.

It will be seen from Fig. 7 that the electrons must be injected into the tube with an accurately determined radial momentum and angular position, in order that

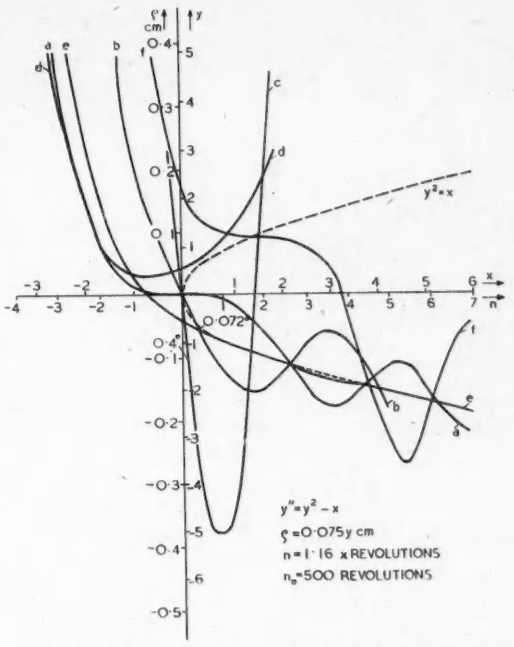


Fig. 7. **Orbits of electrons which are being injected into a potential groove in process of formation.**

they may be captured in the potential groove. The allowable tolerance in width for the angle of injection is only 0·16 deg. in the example mentioned ($B_g = 0{\cdot}9B_0$, $U = 17$ V, $U = 10^4$ V, one full swing in the potential groove corresponding to approximately $3^1/_2$ revolutions) and this would therefore correspond to an injection efficiency of about 4 per cent in the case of a beam with a width of 4 degrees. The width tolerance of the injection angle can be calculated from the relative variation of the depth ΔV_{stab} of the potential groove during the time $\Delta\tau$ of one complete oscillation of the electrons in the groove. From eqs. (46) and (51) the tolerance in width $\Delta\chi$ at the instant τ_1 is

$$\Delta\chi = (\delta\,\Delta V_{\text{stab}}/V_{\text{stab}})^{1/2}, \text{ or}$$

$$\Delta\chi = \sqrt{12a}\left(\frac{p_{\varphi_0}}{3ea\,R_g^3\,B_g}\right)^{3/4}\frac{1}{1+\tau_1}\sqrt{\sqrt{\frac{1}{1+\tau_1}}\,(1+(2\,\tau_1/3))}\;\Delta\,\tau^{1/2} \quad .. \quad (52)$$

where $\tau = \Delta t/t_a = 2\,\pi\,n\,R_g/(v_g\,t_a)$

$$= 2\,\pi\,n\,R_g\Big/\left[4\,\pi\,R_g\,\frac{R_g\,B_g\,U}{R_0\,B_0\,\Delta U}\right] = \frac{B_0\,\Delta U\,n}{B_g\,2U}, \text{ and}$$

$$\Delta\tau = \frac{B_0\,\Delta\,U\,\Delta n}{B_g\,2U_a(1+\tau_1)}$$

When $\tau_1 = 0$ (inflexion point of the potential curve), eq. (52) gives

$$\Delta\chi = \sqrt{12a}\left(\frac{p_{\varphi_0}}{3\,e\,a\,R_g^3\,B_g}\right)^{3/4}\cdot\Delta\,\tau^{3/4} \quad .. \quad (52a)$$

Fig. 8 shows the increase in width ρ_e and the increase in depth V_{stab} of the potential groove, when the initial momentum p_{φ_0} is varied from its maximum value to zero. The figure also gives the variation of the width

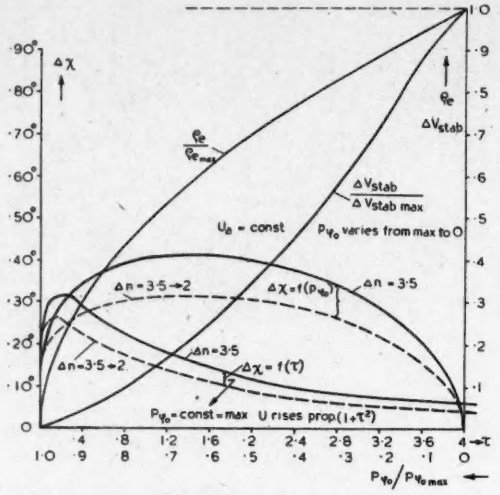


Fig. 8. **Increase of depth and width of a potential groove with decreasing initial momentum p_{φ_0}. Variation of the injection tolerance angle with decreasing initial momentum and increasing time.**

tolerance $\Delta\chi$ for a constant oscillation number $n = 3{\cdot}5$, as well as a curve which takes account of a gradual variation of the number of oscillations from 3·5 to 2·0. This last curve shows that the width tolerance reaches its maximum when the depth of the groove is equal to about 1/5 of its maximum value.

The electrons in the potential groove perform elliptic oscillations if the adiabatic variation of the groove is not taken into account, and the oscillations are determined by the differential equation

$$\ddot{\rho} = \frac{e^2}{m^3} 3\,a\,R_g^2\,B_g^2\,\rho^2$$

The oscillation period will depend on the amplitude of oscillation, and for large amplitudes it increases to 2·2 times the value of a corresponding harmonic oscillation based on a linear force curve.

It is apparent from the foregoing considerations that electron injection is in most cases possible only with a very low injection efficiency. This holds true whether we locate the source of electrons (or the deflecting electrodes of this source) inside or outside the potential groove. Furthermore, to introduce an appreciable charge into the tube, the following points must be borne in mind:—

1. If several potential grooves of different depths exist simultaneously, the space-charge fields of the electrons in the deeper grooves will force the electrons out of the shallower grooves. To maintain a large charge in the tube, all the electrons will therefore have to be provided with the same potential curve, that is, p_{φ_0} will have to remain constant during the charging, and therefore the charging voltage must be raised gradually so as to correspond to the increasing electron voltage.

2. The adiabatic damping of the oscillations will be effective if the resultant depth of the filled groove is not reduced during charging. If charging occurs during the time interval from τ_1 to τ_2, then the maximum charge that can be absorbed will be determined by $V_{stab}\ (\tau_1) - V_{stab}\ (\tau_2)$. The depth of the potential groove is obtained from eqs. (46) and (51):

$$\Delta V_{stab} = \frac{e}{m}\,4a\,R_g^2\,B_g^2\left(\frac{p_{\varphi_0}}{3\,ea\,R_g^3\,B_g}\,\frac{\tau}{1+\tau}\right)^{3/2}(1+\tau)^2$$

where the factor $(1+\tau)^2$ is responsible for the adiabatic oscillation. To introduce a larger charge the charging time $\tau_2 - \tau_1$ must therefore be comparatively long relative to τ_1.

3. As the charging capacity of the potential groove during the charging period grows from zero upwards, the maximum charging current must be considerably larger (about 200 per cent) than the mean charging current.

To generate a charging voltage which will be suitable for the electron potentials considered, one could, for instance, utilize the transient switch-in phenomena of an oscillatory circuit and take the charging voltage from the circuit capacitor. But to raise the injection efficiency, to reduce the charge current, and also the charging time, a new and improved injection method was proposed by Wideröe in 1943. According to this method, a rapidly varying auxiliary field is superimposed upon the magnetic guiding field, and in this way one can obtain the desired rapid variation of the potential curve and hence also a high injection efficiency.

If ΔB is the auxiliary guiding field, then the differential equation for correct-phase electrons will be

$$\ddot{\rho} = \frac{v^2}{R}\left[1 - \frac{e\,R}{mv}(B + \Delta B)\right] = \frac{v^2}{R}\left[1 - \frac{R\,(B+\Delta B)}{R_0\,B_0}\right] \qquad (54)$$

From eqs. (38a) and (46) we have:

$$RB = R_g\,B_g\,(1 - 3\,a\,R_g\,\rho^2).$$

By choosing the auxiliary field B, which is to decrease with a time-constant T, so large that, when $\rho = 0$ and $t = t_a$, the radial acceleration $\ddot{\rho}$ will be equal to zero (so that the equilibrium orbit will be displaced from R_0 towards R_g), we obtain:

$$\Delta B = -\frac{R_g\,B_g - R_0\,B_0}{R_g}\exp(-\Delta t/T) \qquad .. \qquad (55)$$

hence $$\ddot{\rho} = \frac{e^2}{m^2}\,R_0\,B_0\left[3\,a\,R_g\,B_g\,\rho^2 - \frac{R_g\,B_g - R_0\,B_0}{R_g}\left(1 - e^{-\Delta t/T}\right)\right]$$

or $$\ddot{\rho} \cong \frac{e^2}{m^2}\,R_g\,B_g\left[3\,a\,R_g\,B_g\,\rho^2 - \frac{R_g\,B_g - R_0\,B_0}{R_g}\,\frac{\Delta t}{T}\right] \qquad (56)$$

Comparing this expression with that of eq. (48), it will be seen that the time-dependent term has been increased by a factor

$$F = \frac{R_g\,B_g - R_0\,B_0}{P_{\varphi_0}/eR_g}\,\frac{t_a}{T} \qquad .. \qquad .. \qquad (57)$$

In accordance with eqs. (49a) and (49b), this corresponds to an increase in the angular tolerance, and therefore in injection efficiency, equal to $F^{3/5}$. For the example already given, a time constant $T = t_a/20$ would give an increase in injection efficiency of $(24{\cdot}4)^{3/5} = 6{\cdot}8$, and this shows that, by means of an auxiliary field, the injection efficiency can be raised to any desired value. Eqs. (56) and (57) hold only for small values of Δt, i.e. as long as

$$1 - \exp(-\Delta t/T) \cong -\Delta t/T.$$

(To be continued).

DESIGN AND WORKSHOP

A Digest of articles of interest to Works Managers, Planning and Production Engineers, Superintendents, Jig and Tool Designers, Inspectors, Time Study Engineers, and all those concerned with the practical aspects of Engineering.

Aluminium-Copper Cable Connections for Outdoor Stations

(From *Revue de l'Aluminium, France*, Vol. 25, No. 141, February, 1948, p. 65, 3 illustrations.)

CABLES for electric traction are usually connected to outdoor disconnecting switches, transformers or protecting gear the terminals of which are designed to receive copper cable connections. If aluminium or aluminium-steel wires are used* special aluminium-copper unions must be employed. The design of these unions must (1) allow for the difference in metal expansion due to outdoor temperature changes; (2) prevent ingress of moisture which, combined with the electric current, would set up electrolytic cells and rapidly corrode the contact faces; and (3) provide adequate contact area and reliable elastic contact pressure for the safe transmission of heavy currents without overheating. Up to now, the generally accepted design, Fig. 1, made use of bakelite insulating washers

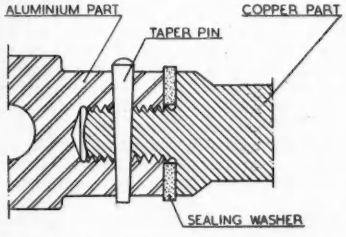


Fig. 1. **Old design of cable connection.**

which separated the dissimilar metals and prevented the ingress of moisture; of screw threads for joining and to provide contact area; and of a taper pin for locking. However, the contact area is insufficient for heavy currents and the contact pressure is not elastic, and is therefore unreliable.

With a view to developing a better and more reliable connection, research was undertaken on behalf of the French Railways and a new design has been evolved, Figs. 2 and 3, in which the aluminium cable is welded into a socket of the union and flat contact faces are provided. The type shown here has an overall length of 5½ inches and is used to connect isolating switches with cables of 0·275 sq. in. cross-sectional area carrying a current of 600 amps. Copper clamp *A* is bolted to the cylindrical terminals *B* as shown. A ¾ inch cadmium plated steel stud *G*, screwed into *A*, holds the aluminium socket *E* which is pressed against bimetal washer *L* and clamp *A* by the action of nut *F* and one or more Belleville spring washers *I*. The annular space around nut *F* is filled with a chemically neutral, thick grease which prevents oxidation and lubricates the spring washers. Aluminium cover *M* retains the grease. The aluminium-copper bimetal washer *L* prevents the ingress of moisture, effectively separates the two dissimilar metals, and provides sufficient contact area so that the electric current density is less than 4·6 amps per sq. in. The spring washers exert an elastic pressure of not less than 1400 psi. The aluminium cable is first introduced into socket *J* of aluminium part *E*, then the single cable strands are welded together at the end, and the lump *K* thus formed is welded to the socket *J*. An oxy-acetylene flame and non-corrosive fluxes are used for welding.

Another type, of 9¾ inches overall length and for aluminium cables of 0·63 sq. in. cross-section is based on the same principle. Copper clamp *A* of this type ends in a flat copper bar, 2 by ⅝ inches in cross-section, and is bolted directly to the terminals. For the better distribution of contact pressure, three ½ inch cadmium plated steel studs are arranged inside the aluminium socket.

* See THE ENGINEERS' DIGEST, May, 1948, p. 147.

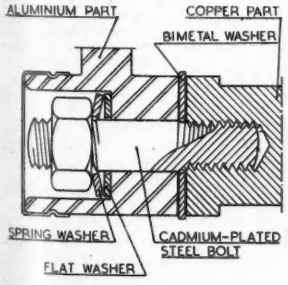


Fig. 2. **New design of cable connection.**

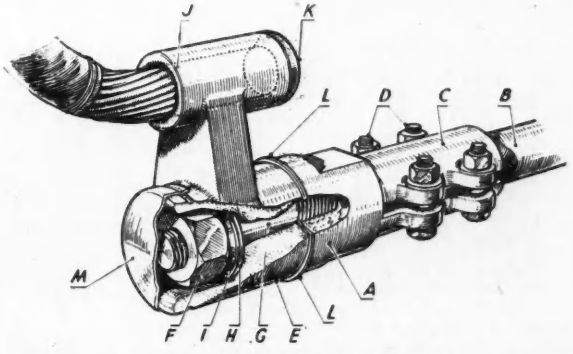


Fig. 3. **View of entire cable connection for cables of 0·275 sq. in. cross-section.**

Economy Factors in Carbide Milling

By A. O. SCHMIDT. (From *The Iron Age, U.S.A.*, Vol. 161, No. 22, May 27, 1948, pp. 72-77, 12 illustrations.)

IN high-speed milling with carbides there are many important factors, often overlooked or easily ignored. Rigidity of an entire set-up, especially in holding the workpiece, and reduction of loading and unloading time are important considerations which can be taken care of in the design of the holding device or fixture, assuming that a machine of sufficient capacity is employed. Better cutter life, accuracy of work, and surface finish are obtainable if the work is held securely by means of fixtures.

Frequently, shops which do not produce parts in large lots cannot justify the expense of special fixtures which would otherwise be desirable. In such cases, a milling machine table, also called chucking table, specially designed to permit rigid holding and quick handling of a great variety of shapes will prove profitable.

Because of the decrease in time per piece possible with carbide milling, cutters may wear out in a shorter time than would high-speed steel cutters although, of course, they will produce more pieces in that time. A question which naturally arises, therefore, is how many passes can be made with a carbide cutter. This is difficult to answer when a particular kind of workpiece must be machined for the first time, or a new cutter design, a different tool material, machine tool or fixture has to be used.

Generally, after a number of trial runs the experienced shop man will be able to determine feeds and speeds which will permit near optimum production and also will be able to eliminate detrimental factors. This procedure will take varying amounts of time depending upon the conditions of the job. However, more or less constant improvement can usually be expected the more closely the job is attended. The following description of a particular milling operation in the shop serves to illustrate what can be done to improve performance after feeds and speeds have been settled upon. It also provides a typical answer to the question of how many passes a cutter can make.

A large steel block, Fig. 1, annealed, 220 Brinell hardness was to be reduced in size. Ultimately this required the removal of about 1½ tons of chips. A boring machine with a table large enough to accommodate this block and with a 15-hp spindle drive motor was selected. The cutters used were 8-in CSM cutters with ten inserted solid carbide blades. The width of cut of 3-in depth of cut of 0·200-in, feed of 13 ipm, and cutting speed of 310 fpm required between 14·5 and 17·5 hp depending on cutter conditions. Shortly before the cutters were removed for regrinding they usually were drawing higher horse-power due to blade wear.

Fig. 2 graphically illustrates how the number of passes per cutter varied over a period of three days. The average for all six shifts was ten passes per cutter. During the first two shifts the cutters failed more often by chipping or fracture and only 107 passes were completed with 12 cutters. Since this particular machine had seldom been used for such heavy work, adjustments had to be made to correct looseness of clutches and play in shafts and in table movement which became evident during the first shift. During the third and fourth shifts fewer stoppages occurred since the machine was then better adjusted for this operation. F: een cutters completed 148 passes. Further improven..nt can be

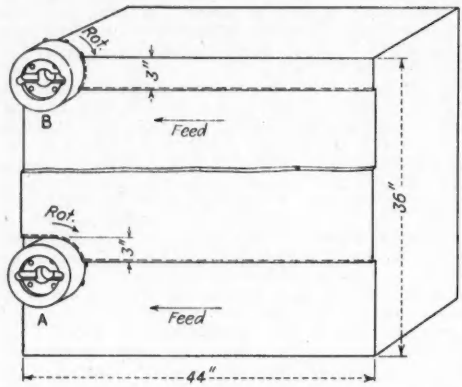


Fig. 1. Diagrammatic view of a block of steel which was milled with identical cutters through six work shifts.

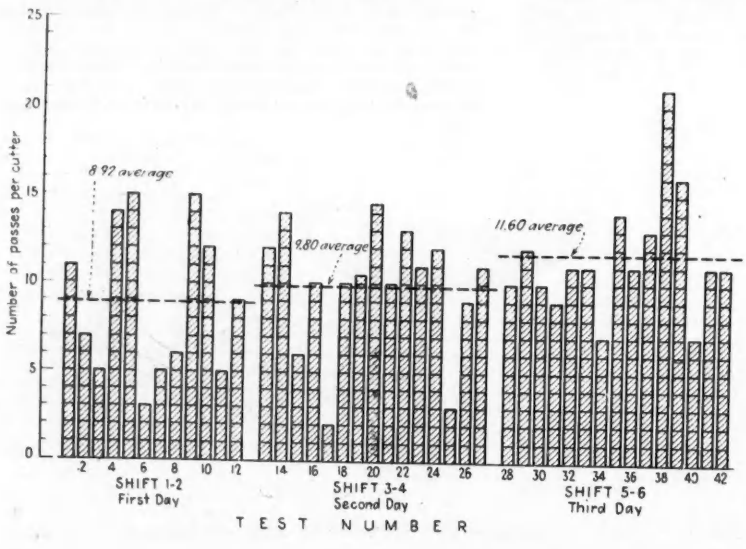


Fig. 2. This graph shows the number of passes made with each cutter in three successive days.

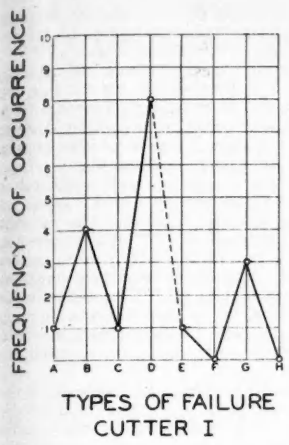


Fig. 3

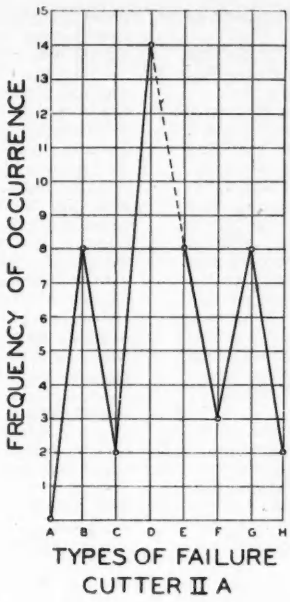


Fig. 4

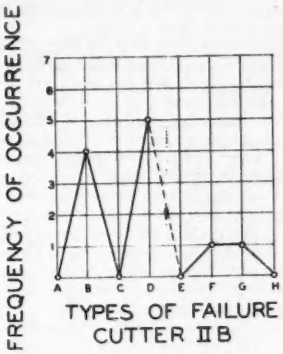


Fig. 5

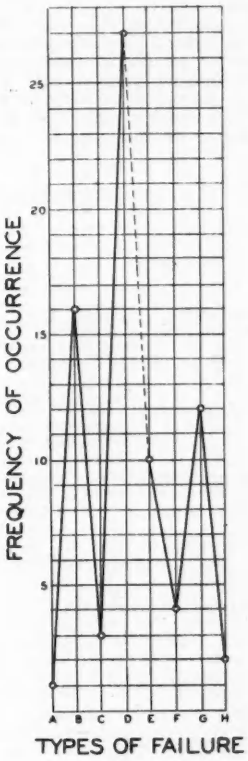


Fig. 6

Fig. 3. **Graphical analysis showing the type and frequency of wear and failure of 26 face milling cutters of Type I, which was most economical in overall performance and machined and average of 154 pieces.**

Type I is a 10-in CSM face milling cutter having 12 solid sintered carbid cutting blades set at a radial rake angle of 15° positive. A negative rake angle face of 10°, 0·25-in wide, is provided at the peripheral cutting edge. The wedge is in back of the blade to provide uninterrupted chip accommodation space.

Fig. 4. **Graphical analysis showing the type and frequency of wear and failure of 38 face milling cutters of Type IIA. More undesirable type of failure and excessive wear occurred with this cutter. An average of 98 pieces per cutter were finished.**

Type IIA is a 10-in face milling cutter with 12 solid sintered carbide cutting blades set at a radial rake angle of 7° negative and an axial rake angle of 7° negative.

Fig. 5. **Graphical analysis showing the type and frequency of wear and failure of 10 face milling cutters of Type IIB. Although the performance was satisfactory, the replacement of tips was frequently necessary and more expensive than with the other cutters. An average of 147 pieces per cutter were finished.**

Type IIB is a 10-in face milling cutter, with 12 brazed sintered carbide cutting tips, rake angles as for Type IIA.

Fig. 6. **Graphical analysis of the wear and failure of all three types of cutters. They were used on the same job and machine.**

noted in the fifth and sixth shifts with 15 cutters completing 174 passes. General performance undoubtedly would have been improved had a heavy milling machine, instead of a boring machine, been available.

When the cutter was milling as in position *A* of Fig. 1, the wattmeter indicated 15 hp but only 12 hp in position *B*, the decrease being due to the greater efficiency inherent in the formation of shorter and thicker chips.

The cutters used in these tests were reground by experienced operators and had a 6° negative radial rake angle imposed at the cutting edge of a carbide blade positioned in the cutter body at a 15° positive-radial-rake angle.[1] Quite uniform cutter conditions were assured when cutters were returned from the grinding department since all wear marks on the blades were eliminated in each regrinding. Cutters were removed when they showed $\frac{1}{32}$-in wear on the peripheral relief face. The low number of passes on the first and second day was accompanied by cracking or chipping of the carbide blades. This type of failure can occur because of hard spots in the workpiece or momentary stalling of the

cutter in the work, resulting from a loose clutch or an unexpectedly heavy chip.

The results plotted in Fig. 2 represent production data under shop conditions as influenced by the machine, workpiece, and, last but not least, a good operator.

The graphs in Figs. 3, 4, 5 and 6 show the frequency of the various types of wear and failure of carbide cutters. The operation analysed consisted of face milling alloy steel wing hinges.[2] Cutting speeds, cutter diameter, number of blades, and grade of carbide resulting in good operation had been previously established. The particular type of wear and failure of carbide cutters used in each run was recorded. Three different designs of cutters were used. These cutters were alike in size, number of blades, grade of carbide, relief angles, as well as true rake, the distinguishing feature of each type of cutter being in the manner in which the carbide material was attached. Cutters with solid carbide blades wedged at either a positive or negative angle and with tips brazed directly to the body were tested. Altogether, 74 face mills of 10-in dia. each having 12 teeth were checked.

The machine operator had been instructed to remove the cutter for resharpening as soon as the area of wear on the peripheral relief face became $\frac{1}{32}$-in wide ($\frac{1}{32}$-in peripheral wear). Among other reasons, this criterion was chosen because it is easily visible and can be measured with a scale. If this type of wear prevails, it generally indicates that good cutting conditions exist.

TABLE I. TYPES OF WEAR OR TOOL FAILURE OCCURING IN THREE DIFFERENT MILLING CUTTERS.

Types of Wear or Failure.		Cutters* Type I	Type IIA	Type IIB
Less Objectionable Types of Wear	A—Excessive Peripheral Wear	1	0	0
	B—Corner Wear	4	8	4
	C—Face Wear	1	2	0
	D—1/32-in Peripheral Wear	8	14	5
More Objectionable Types of Wear	E—Crater	1	8	0
	F—Excessive Corner Wear	0	3	1
	G—Slight Chipping	3	8	1
	H—Breakages of Some Blades	0	2	0

*Numbers shown indicate how often the particular type of wear or failure occurred in the cutter tested.

Other types of wear and failure occur quite frequently and those observed in these operations are listed in Table I. The types of wear designated as A, B, C, and D are the more desirable since they can be rather easily eliminated in a relatively short time by regrinding. Types designated E, F, G, and H are objectionable, because regrinding is more difficult and time consuming, and occasional replacement of carbide tips is required. If the latter types of failure occur too frequently it may be necessary to change the cutting speed, feed, depth of cut, grade of carbide, work-holding device, or cutter design; or shift the job to another machine of greater power and rigidity. The frequency of occurrence of these various kinds of wear and failure in the three different cutter designs is plotted in Figs. 3, 4 and 5. From this analysis it was possible to determine the type of cutter most economical in regard to carbide wear and regrinding and resetting of blades.

A composite presentation of Figs. 3, 4, and 5 is shown in Fig. 6. For a given operation this diagram indicates conditions may be such that there is unobjectionable cutter wear, but that a certain amount of objectionable failure and excessive wear will also occur. If wear and failure of the kinds designated by E, F, G, and H of Table I result too often, the entire operation may become uneconomical. High cutting speeds, excessively heavy feeds, hardness of workpiece, incorrect grade of carbide, or an overloaded machine are the main contributing factors to objectionable types of failure, especially in milling high-strength alloy steels.

The most commonly used methods of holding sintered carbides in a milling cutter are brazing the tips directly to the body or attaching removable teeth to the body with various mechanical locking devices. A removal tooth is either a steel blade to which a carbide tip has been brazed or a solid carbide blade. Tests indicate that carbide tips properly brazed to the cutter body are as strong as mechanically held blades. A cutter with tips brazed to the body is often the best selection in certain instances and is usually lower in initial cost. However, in long production runs, cutters with mechanically-held blades will be more economical because resetting the blades in case of breakage or for resharpening is more simple and less time consuming. A hardened steel cutter body with mechanically-held blades is not weakened or damaged by repeated regrinding and brazing.

A typical time study is shown in Table II of the reconditioning of a cutter with tips brazed to the body.

An identical cutter with solid sintered carbide blades wedged into the cutter body at a negative radial rake angle also required 35-min for regrinding, but only 15-min for resetting a blade in case of breakage. After four or five resharpenings all blades must be reset and sharpened, but this requires only $1\frac{1}{2}$ hr.

The CSM cutter with solid sintered carbide blades wedged into the cutter body at a positive radial rake angle required 45 min for regrinding. Resetting a blade in case of breakage takes 15-min. However, since less material has to be removed in regrinding because of this particular blade design, this cutter can be resharpened eight to ten times, before resetting and sharpening which requires $1\frac{1}{2}$ hr, is necessary. Less breakage of single tips is another factor in its favour. In production runs fewer cutter changes with attendant delays will occur and the number of workpieces finished per cutter will be high.

The number of workpieces machined per hour with a particular cutter is of prime importance in determining the economy of a carbide milling operation. Loading and unloading time is often much greater than actual cutting time and therefore much attention must be directed to decreasing this time with proper holding devices or fixtures. Rigidity of both the set-up and the milling machine and adequate power as well as proper choice, use and servicing of cutters will help to "cut cost."

TABLE II. A TYPICAL TIME STUDY OF THE RECONDITIONING OF A 12 TOOTH FACE MILL WITH TIPS BRAZED TO THE CUTTER BODY, EXCLUDING MOVING TIME BETWEEN SHOP DEPARTMENTS.*

Operation No.	Operation.	Time, Minutes
1	Selecting, Grinding and Delivering New Tips to Brazing Dept.	10
2	Removing Old Tips and Brazing New Tips	54
3	Sandblasting Cutter	8
4	Grinding Faces of Teeth	45
5	Grinding Periphery of Cutter Teeth	48
6	Finish Grinding Relief and Clearances	45
	Total Time	.. 210 min or 3½ hr

*If tips do not break, the cutter can be resharpened in 35-min, but can be resharpened only about three to five times before it must go through the reconditioning procedure outlined. In the event that only one tooth is broken, a new tip must be brazed on by following the outlined sequence, and to put the cutter into working condition again, about 100-min is required.

REFERENCES

(1) "Radial Rake Angles in Face Milling," by J. B. Armitage and A. O. Schmidt, *Mechanical Engineering*, Vol. 67, No. 8, August, 1945, pp. 507 to 510.

(2) "Data on Production Runs with Carbide Milling Cutters," by A. O. Schmidt, *The Iron Aage*, Vol. 156, No. 21, pp. 54 to 57, Nov. 22, 1945.

Dimensional Analysis of Engineering Designs

(From a Ministry of Supply publication* prepared by *Inter-Services Committee for Dimensioning and Tolerancing of Drawings.*)

INTRODUCTION

In 1944, the Ministry of Production set up an Inter-Services Committee "(a) to establish for armament work a basic system for tolerancing and dimensioning drawings of stores. Gauges, jigs and fixtures of various degrees of precision; (b) to make recommendations as to how the principles and practice so established can be made known to and applied by the Engineering Industry." The Committee decided first to establish a series of basic principles to guide designers in determining appropriate basic sizes, tolerances, allowances for components for interchangeable production, then to show applications of these principles to various types of dimensional problems. Whilst these problems are of a fundamentally geometrical character, practical aspects of tooling and gauging are kept in mind. Although requirements of armament production are more stringent than those of peace-time industry where the designer knows the production facilities available and can consult with shop personnel, the principles for the analysis of design are equally applicable to both.

TEN BASIC PRINCIPLES OF DIMENSIONAL ANALYSIS

Principle I. Requirements relating to interchangeability should be determined from considerations of simplicity and economy of production, inspection and servicing.

Complete interchangeability of all components simplifies assembly and servicing, facilitates substitution of spare parts. Dimensional requirements, however, are rigid and insistence on interchangeability of every detail may become uneconomical. The "*unit assembly*" may then be chosen in which no part is required to be interchangeable but the whole unit can be supplied as a spare part for the finished product. Details of an interchangeable assembly are inspected for dimensional accuracy, and the unit assembly is submitted to a functional test for correct functioning. Dimensional checks may be additional if no proper functioning test can be devised. Intermediate possibilities are:—interchangeable assemblies with unit sub-assemblies or unit assemblies in which, nevertheless, certain special details are interchangeable. Fig. 1 shows a unit sub-assembly, finish machined after assembly so that the

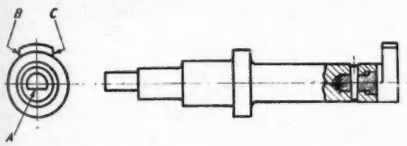


Fig. 1. **Example of details requiring to be assembled before final machining of A to ensure its correct angular relationship with faces B and C.**

details of this unit are not interchangeable although the whole unit may form an interchangeable part of a larger assembly. Welded articles can be regarded as unit sub-assemblies. *Selective assemblies*, Fig. 2, avoid uneconomically close tolerances. In Fig. 2 all parts are interchangeable, only washer *D* is selected to give an end play *A* within the desired limits. Several washers of different, closely controlled, thicknesses will be supplied as spare parts with the assembly. Designers should find the most economical compromise, the guiding rule being:—maximum interchangeability of

*Published in 1948 by H.M. Stationery Office, 429 Oxford Street, London, W.1. Price 7/6 net.

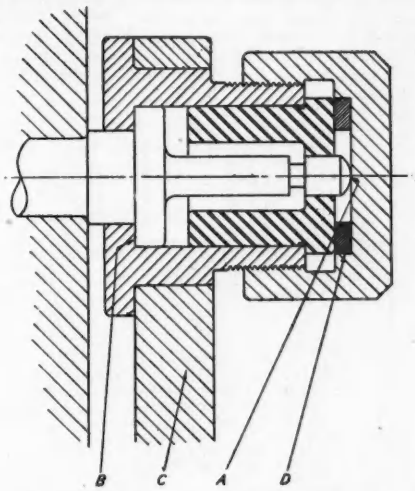


Fig. 2. **Example of design using selective assembly of washer D to obtain close control of end play at A.**

details with minimum costs and maximum simplicity.

Principle II. Standard sizes and parts should be used wherever possible.

Standards aim at overall economy of time and effort in producing and servicing. They reduce detail design work, production and inspection costs, and the number of types of components needed to maintain a given range of equipment. Standards often show the best known way of getting something done and the guiding rule for the designer should be that he must adopt the appropriate standard unless he can prove that there is a definite advantage in departing from it. *National standards* embrace the whole general engineering industry, *trade standards*, although more limited, may still be too wide, and *local standards* should be selected to suit the limited field of one firm or department. Standards should not just "grow" but follow a definite general policy of economic design. A special standards section to promote standardizing activities and issue standards to the design office would provide the best means of co-ordinating the local standards. For dimensional requirements, local standards should aim at establishing :—

(a) general standards (such as preferred sizes, limits and fits, screw threads, etc.);
(b) common engineering component standards (such as bolts, nuts, studs, washers, pins, etc.);
(c) specialized design standards, i.e.
    (c1) finished standard components which can be assembled into different assemblies (e.g. one crank shaft suitable for different types or sizes of engines); (c2) not completely finished components which may be processed or machined to produce slightly different finished components.

Standards under (a) reduce tooling and gauging costs. A series of preferred basic sizes should first be selected, then a limits and fits scheme for plain and screwed holes and shafts, gaps and widths as suited to local needs. A "first preference" should be indicated in the selected standards series and, since no series can

cover all requirements, a desirable method for selecting "specials" be recommended.

Fig. 3 is typical for a range of assemblies where dimension *A* varies with *B*. Features requiring to be altered are dimensions *C* with *D*, *E*, tooth pitch with dimension *F*, and the cam profile. If the design was planned for a whole anticipated range of sizes, then optimum values could probably be chosen for *C*, *D*, *E*, *F* and the pitch, so that only the cam profile would not be standard.

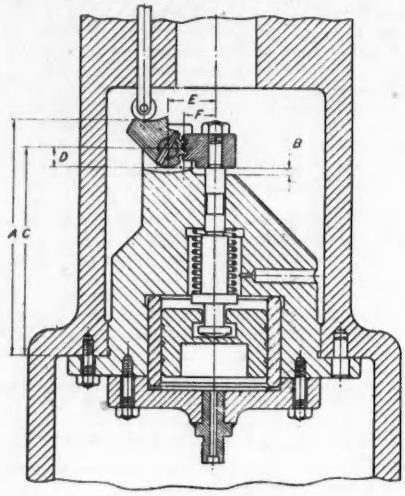


Fig. 3. **Example illustrating typical design of a range of assemblies.**

Principle III. Fits between mating parts should be arranged in general on the hole basis. The shaft basis should only be used where the hole basis is unsuitable.

The hole basis provides one standard hole size whilst the shaft diameters vary according to the fit required. The shaft basis, with one standard shaft size and holes to suit the fit, normally leads to higher tooling costs but may be advantageous where it is more economical to remove the need for machining the shaft by using bar stock.

Principle IV. Features of details of an assembly should be carefully studied and design sizes first assigned to those dimensions which affect assembly or functioning or both.

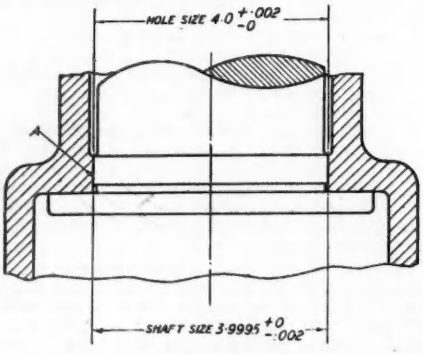


Fig. 4. **Example of close clearance fit.**

NOTE.—Systematic analysis according to this principle automatically establishes the functioning data.

Principle V. Where variations from the desired assembly and/or functioning requirements can be permitted only in one direction, appropriate *unilateral* tolerances should be assigned to the relevant design sizes. Where variations can be permitted in both directions *bilateral* tolerances should be used.

Observance of these principles will lead to simplification of dimensioning problems, to critical examination and perhaps improvement of a design. Designers should first forget all about tolerancing and determine the design sizes for dimensions, and then only apply principle 5 to find the direction and magnitude of permissible tolerances. Fig. 4 illustrates a clearance fit between spigot and hole. The basic size of 4·0-in has been chosen as the theoretical size on which the design is based. The hole basis for the fit has been selected, and the design size of the hole is, therefore, the basic size, 4·0-in. A small allowance is needed for assembly, so the designer allows a minimum clearance of 0·0005-in, and the design size for the spigot is thus 3·9995-in. Since perfect workmanship cannot be obtained, tolerances must be assigned to these design sizes. Variations from the design sizes are only permitted in one direction as the minimum clearance must be kept, and unilateral tolerances are chosen as shown. In Fig. 5 dimension *F* is required to be

$$4{\cdot}25\text{-in} \begin{matrix} + \\ - \end{matrix} \begin{matrix} {\cdot}01\text{-in} \\ 0 \end{matrix}$$

when the piston is driven hard up against the cap. From the design sizes which establish this desired function it follows that $F = A - B - C + D + E$ and that, if the required tolerance for *F* is to be adhered to,

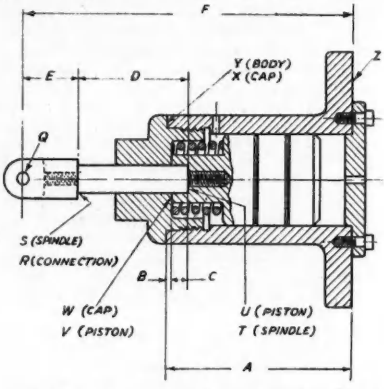


Fig. 5. **Choice of primary dimensions from analysis of functioning surfaces.**

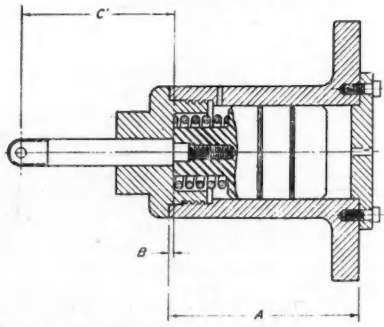


Fig. 6. **Illustrating the design alterations to Fig. 5 to reduce the number of primary dimensions.**

each of the five design dimensions must have a unilateral tolerance of not more than 0·002-in. If the design is slightly altered as in Fig. 6, then $F = A - B + C$; only three design dimensions are used to establish the desired function and the tolerances for each of these can be larger, with consequent reduction of production costs. Thus, logical analysis may lead to design alterations and production improvement.

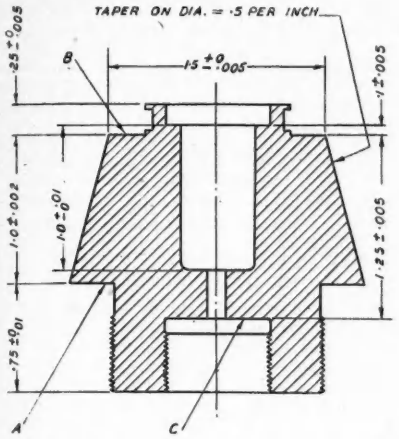


Fig. 7. **Illustrating design sizes and tolerances as found by applying Principles 4 and 5.**

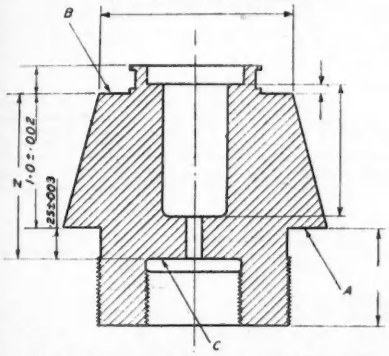


Fig. 8. **Illustrating alterations in dimensioning for component, Fig. 7, consequent upon change in datum.**

Principle VI. Functioning data established by applying principle 4 should be examined for suitability as economical production and inspection data. If they are unsuitable, other data should be chosen.

Fig. 7 shows the design of a component correctly dimensioned according to principles 4 and 5. Face *B* is the main functioning datum to which dimensions for other faces refer. If, however, face *C* has to be machined in direct relation to *B*, i.e. in one machining operation, then difficulties of clamping the component in the lathe and of gauging distance $B - C$ would arise. Dimensions should therefore be altered as shown in Fig. 8 where face *A* is selected as datum for *C* and the distance $B - C$ subdivided into $B - A + A - C$, with a corresponding *reduction of tolerances* for each of these dimensions. The threads, faces *A* and *C* can then be finish-machined in a first operation, the component screwed onto an adaptor, hard up against face *A*, and face *B* machined in a second operation. Inspection is also facilitated. This principle should be applied with care and doubtful cases be discussed with the planners.

Principle VII. Tolerances should be so assigned that no dimension is subject to redundant limits.

One of the three dimensions *A*, *B*, and *C* in Fig. 9 should be omitted. Suppose *A* and *B* are to be 1·0 ± ·002-in each, then *C* would be 2·0 ± ·004-in. If the

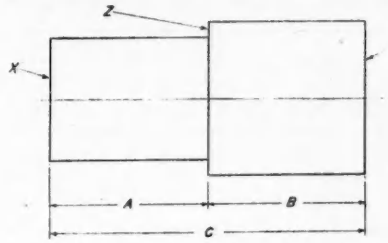


Fig. 9. **Illustrating redundant dimensions.**

dimension of *C* is given on the drawing, the workshop could work to dimensions *A* and *C* only, leaving *B* to be 1·0 ± ·006-in which would contradict the intentions of the designer. If dimension *C* is given as overall length for the convenience of the planning department, it should be untoleranced and clearly marked as being a reference dimension only.

Principle VIII. Geometrical layout and method of tolerancing of complex work should be scrutinized, and if necessary modified, to suit practical and inspection requirements.

Manufacturing process or inspection method may determine a correct dimensioning procedure. The size of the tapered hole Fig. 10 depends on how far the

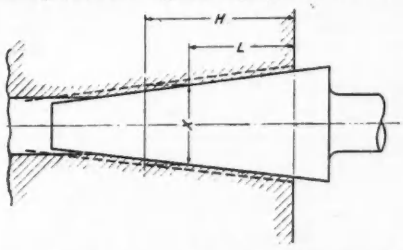


Fig. 10. **Illustrating method of dimensioning tapered work obtained from analysis of probable manufacturing errors when using taper reamers.**

tapered reamer has been inserted. This consideration leads at once to the dimensioning system shown in the sketch, with the distance of basic diameter *X* from one end face toleranced between *H* and *L*. Full and dotted lines show a parallel band of tolerance within which the hole may vary. A simple system of taper-plug gauges will automatically permit this parallel band of tolerance for the work. In specially complicated cases and as a last resort, dimensional requirements of a component can be stated by reference to the drawing of a suitable gauge issued to check the work. The gauge drawing is then part of the component drawing and the two must be issued together. Designers should always aim at stating quite definitely, clearly and directly on the drawing the essential requirements that must be observed for satisfactory work even if this leads to apparently unusual methods of dimensioning.

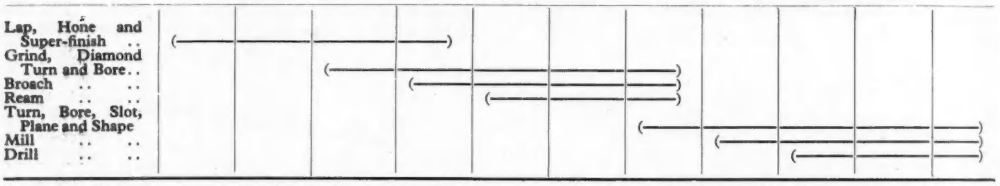
Principle IX. Tolerances should be assigned to features for position, concentricity, symmetry, flatness, roundness, squareness, and parallelism when control of these elements is important for correct assembly or functioning or both.

Normally, only dimensional tolerances are given on drawings and trouble may arise during assembly if

TABLE I. TOLERANCES SUITABLE FOR COMMON MACHINING PROCESSES.

Range of Basic Sizes (inches)		Grade of Workmanship										
		1	2	3	4	5	6	7	8	9	10	11
Above	Up to and including	Unit of tolerance — ·001 in.										
0	·6	·06	·1	·2	·3	·4	·6	1·0	1·5	2·5	4·0	6·0
·6	·7	·06	·1	·2	·3	·5	·8	1·2	2·0	3·0	5·0	8·0
·7	1·1	·06	·1	·2	·3	·5	1·0	1·5	2·5	4·0	6·0	10·0
1·1	1·6	·06	·1	·2	·3	·6	1·2	2·0	3·0	5·0	8·0	12·0
1·6	3·1	·06	·1	·2	·4	·8	1·5	2·5	4·0	6·0	10·0	15·0
3·1	4·1	·06	·1	·2	·5	1·0	2·0	3·0	5·0	8·0	12·0	20·0
4·1	5·1	·1	·2	·3	·6	1·0	2·0	3·0	5·0	8·0	12·0	20·0
5·1	6·1	·1	·2	·4	·6	1·2	2·5	4·0	6·0	10·0	15·0	25·0
6·1	8·1	·1	·2	·4	·8	1·5	3·0	5·0	8·0	12·0	20·0	30·0
8·1	10·1	·2	·3	·6	1·0	1·5	3·0	5·0	8·0	12·0	20·0	30·0
10·1	12·1	·2	·4	·6	1·2	2·0	4·0	6·0	10·0	15·0	25·0	40·0
12·1	15·1	·3	·5	·8	1·5	2·5	4·0	6·0	10·0	15·0	25·0	40·0
15·1	20·1	·4	·6	1·0	2·0	3·0	5·0	8·0	12·0	20·0	30·0	50·0
20·1	25·1	·5	·8	1·2	2·0	3·0	5·0	8·0	12·0	20·0	30·0	50·0
25·1	30·1	·6	1·0	1·5	2·5	4·0	6·0	10·0	15·0	25·0	40·0	60·0
30·1	40·1	·8	1·2	2·0	2·5	4·0	6·0	10·0	15·0	25·0	40·0	60·0
40·1	50·1	1·0	1·5	2·5	3·0	5·0	8·0	12·0	20·0	30·0	50·0	80·0
50·1	60·1	1·2	2·0	2·5	2·5	5·0	8·0	12·0	20·0	30·0	50·0	80·0

Related Machining Processes



NOTES.— 1. Grades 1 to 5 are intended primarily for gauge tolerances but may be used for fine work tolerances where necessary. It is recommended that direct measurement should be used to check shafts with tolerances finer than Grade 6, and holes with tolerances finer than Grade 5.

2. Grades 3 to 8 are those considered appropriate to jigs, fixtures and tools.

3. Table 1 is intended for use with gaps, widths and lengths as well as for cylindrical holes and shafts.

4. The tolerances given in the above table are for guidance only, and are not to be regarded as a recommended basis for a limits and fits system.

dimensional tolerances are wholly absorbed by errors in other features, e.g. position of holes or concentricity of cylinders in mating components.*

Principle X. Tolerances should be as wide as is consistent with satisfactory functioning and economical production and inspection. No special difficulties should be imposed without real functional necessity.

NOTE.—Tolerances on jigs, fixtures and cutting tools are approximately 20 to 50 per cent, tolerances on gauges approximately 10 per cent of the corresponding work tolerances.

Tolerances should first be fixed for functioning performance desired, then examined and perhaps altered for ease of production and inspection. Up to a certain point, larger tolerances reduce costs, and Table I gives a series of basic tolerances as a guide for the designer and indicates the accuracy that can normally be expected from standard machining processes. The finest grade given for a particular machining process indicates the smallest tolerance that should be quoted for this process, the coarsest grade indicates the largest tolerance that need be quoted. If finer tolerances are required, production engineers should be consulted about the special difficulties that may be involved. The tolerances shown in Table I apply to features important for assembly or functioning; unimportant features should have larger tolerances.

*Often, the analysis of complex work is not easy, and suitable lines of approach for the designer are suggested in the relevant chapters of the book. Sound methods for correctly grouping and tolerancing important features, like positional work, are developed by analysis of production and inspection methods.

In this Issue—*continued from page* 289

The stepless positive regulation of speeds and forces, within a wide range, at high rotational speeds and power outputs, constitutes a problem often encountered in many fields of mechanical and chemical engineering. It is also of importance in the food, textile, paper and other specialized industries.

Existing hydraulic gears, based on the principle of axial or radial positive displacement pumps, suffer from the drawback that they are limited both by the permissible number of revolutions and by the operating pressure. The article on the "Hydro-Titan" gear published on page 290 shows how Swiss engineers have overcome these limitations by the development of a hydraulic drive, on which patents have been applied for in most countries. The advantages of this drive are its great flexibility, high power output and small space requirements.

Hot-Pressing and Hot-Forging Light Alloys

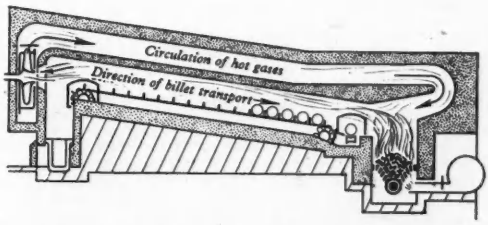
By M. CHARTRON. (From *Revue de l'Aluminium, France*, Vol. 25, No. 141 and 143, pp. 37-43, and 113-122, 27 illustrations.)

I. FORGEABLE LIGHT METALS

FORGED pure aluminium is sometimes needed for its chemical stability, corrosion resistance and its qualities as a sealing material in pipe joints. Aluminium can be forged at any temperature, from below 550° C to room temperature, but its mechanical strength is low at whatever forging temperature is chosen. A test piece of 99·5 per cent aluminium was cut from a bar and (a) forged into a disk in one heat at 500° C, then drop-forged in one heat to final shape, and (b) cold-forged into a disk and stamped to final shape with the same tools. Its mechanical properties after working were :—

	Elast. Limit psi	U.T.S. psi	Elongation per cent
Hot-forged	10,000	13,000	18
Cold-worked ..	14,000	17,000	5

Table I gives the usual wrought light alloys with their properties, and heat treatment and forging temperatures. They are delivered to the forge either in the form of cast ingots or as extruded or rolled bars. They are brittle in the cast state, and, therefore, ingots must be cold-worked before being forged.



easy handling, and they do not generate combustion gases. They heat by radiation from the internal walls, however, and because light alloy surfaces reflect radiation, their rate of heating is too slow for forge purposes, even if fans are used inside the furnace to increase draught. An improvement on this can be obtained by using furnaces based on forced convection (Fig. 1), in which air or gases are heated by coke fires, gas or crude oil burners, or electrically, with automatic conveyance of billets from entry to outlet door, and with automatic furnace temperature control. The salt bath furnace usually contains sodium nitrate heated by gas, oil or electricity, which becomes liquid just below forging

TABLE I. FORGEABLE LIGHT ALLOYS IN GENERAL USE.

Material	Composition in per cent			Mechanical properties			Heat treatment				Forgeable
				Elast. limit	U.T.S.	Elong.	Quench after		Anneal		
	Cu	Mg	others	psi	psi	%	min. at deg. C.		hrs. at deg. C		at deg. C
Duralumin	4	0·5	0·5 Mn ; 0·6 Si	35000	57000	16	20	490–500			440–350
Duralumin FR	4·2	1	0·6 Mn ; 0·15 Si	43000	63000	16	20	490–500			440–350
Hiduminium RR 56 ..	2	0·9	0·7 Si ; 1·3 Ni 0·1 Ti ; 1·3 Fe	50000	63000	8	120	520–530	16	170	480–350
Y-alloy	4	1·5	2 Ni	40000	54000	12		515–520	9	175	480–350
Almasil	—	0·7	1·5 Si	31000	43000	14	20	525–530	3	175	500–300

II. FORGE FURNACES

The open coke fire is still widely used. The end of a long bar is held over the red-hot coke, and its temperature can be judged from the blackening, at 300° C, of a line drawn on the bar with ordinary soap. A well planed soft wood board will discolour when rubbed at the heated end, at temperatures above 300° C. From the colouring, temperatures can be judged to within ± 10° C. Disadvantages are the variations of temperature at the heated end, the need for constant supervision, and the waste of material when heated on the bar. In induced draught furnaces the billets are placed near, but away from, the coke fire and heated partly by radiation, and partly by convection from the air blown through the fire. The furnace temperature near the billets can be measured by using thermo-couples. Disadvantages are inhomogeneous heating of billets, blackening by the gases, and the need for careful fan control because the draught speed determines the rate of temperature increase. Nevertheless, if the necessary precautions are taken, this type of furnace can give satisfaction and is very economical in use. Electric furnaces, as normally used for the heat treatment of light alloy sheet, rivets, and sections have the advantages of accurate temperature control, uniform heating, and temperature, affords the quickest method of heating light alloys. Their high conductivity assures uniform heating of the total cross-section. It has been shown that the centre of a 10-in diameter bar was heated to the required temperature only two minutes after the surface layers had reached it. The disadvantages are high running costs, a long starting-up time for melting the salt bath, and the necessity for cleaning hot billets before working to prevent the bottom die from becoming filled with liquid salt and so to safeguard the operator against burns. This type of furnace, although excellent for heat treatment, is, therefore, not recommended for heating light alloys. Assuming average service conditions and continuous operation of the various types of furnace, their thermal efficiency, and fuel and energy consumption to heat one ton of Duralumin (specific heat 0·23) from room temperature to 450° C are as follows :—

Type of furnace	Av. thermal efficiency cont. run. per cent	Consumption	Calorific value per unit of fuel
Electric furnace	50	245 kWh	—
Forced-convection, heated by coal	15	204 lbs	13,500 BTU/lb
Forced-convection by crude oil ..	20	116 lbs	18.000 BTU/lb
Forced-convection by town gas	20	197 cb yds	10,600 BTU/cb yd

Normally, coal and oil furnaces will give the cheapest service and the electric furnace will be the most ex-

pensive in operation, and even more so when, for 8 hours running per day, its thermal efficiency drops to 33 per cent. Full transport and storage costs, servicing, maintenance, and capital charges must be added, but will not materially affect the overall cost, unless special economic conditions favour the use of an alternative type of furnace.

III. FORGING CONDITIONS FOR LIGHT ALLOYS

A. Forging with Power Hammer

(1) The colour of the light metal does not change when it is heated to forging temperature. The temperature cannot, therefore, be judged by the naked eye and must be measured when billets are taken out of the furnace. This can be done by contact pyrometer, or empirically by touching the hot metal with a piece of paper or a well-dried soft-wood board, or by throwing sawdust on to the metal. The paper or board discolours on contact with the hot metal, the colour varying from straw-yellow to brown according to the temperature of the metal. When sawdust is used, some of it will ignite on contact with the metal, and an experienced operator will be able to estimate the temperature, according to the amount ignited, to within $\pm$ 10° C. The pyrometer is more accurate but its galvanometer is easily damaged by the hammer blows, the effects of which are propagated throughout the building.

(2) If the metal being forged does not lie flat on the anvil, violent shocks are transmitted to the tongs. The operator must, therefore, be experienced, and the power hammer so designed to allow perfectly clear vision of the position of the metal on the anvil. The operator must be able to stop the hammer instantaneously from any position to prevent accidents.

(3) To change the position of the metal being forged a considerable effort, in no relation to the weight of the metal, is required due to the latter sticking to the anvil. Any roughness of the anvil surface will mark the forging and, therefore, the anvil and tup surfaces should be very smooth and lightly lubricated.

(4) The temperature range for forging is as small as 60 to 80° C, therefore the power hammer must be designed to impart a maximum amount of energy in a given time. Operation must be rapid and powerful, and the equipment robust.

B. Drop Forging

Similar remarks apply to forging between die-blocks. The metal does not flow easily in the dies and seems to stick more than when forged between flat surfaces. Operation is more difficult than in the case of drop-forging of mild steel, and the effort required increases very rapidly as the temperature of the work piece falls. The power hammer must not only be powerful and robust but also work accurately. Since the metal does not scale on heating, it can be drop-forged to finish dimensions, subsequent machining being unnecessary.

C. Hot-Pressing

The press works without shocks, but although it can produce considerable deformation of the work piece, the deformation can only take place within a short period owing to the very rapid cooling of the work piece. The actual degree of deformation in one heat may thus be less than that obtained under a power hammer.

(To be continued.)

Recovery of Defective Steel Parts by Electro-Deposition

(From *Machines et Métaux, France*, Vol. 32, No. 357, May, 1948, pp. 161-162, 2 illustrations.)

STEEL parts which have been scrapped by faulty machining can be retrieved by building up with electrolytic deposits of chromium or nickel. Chromium, although easier to deposit, is very hard and brittle whilst nickel has more steel-like qualities and can be made so adherent that even shocks or machining operations do not separate the deposit from the base metal. Nickel also penetrates better into spots not easily accessible, and into intricate shapes of the part to be recovered. Deposition can, in every case, be limited to a certain part of the surface only.

The hardness of nickel deposits can be varied by modifying the process conditions (150 to 400 diamond hardness), the U.T.S. may vary between 60,000 and 150,000 psi and the elongation correspondingly from 30 to 6 per cent. On certain surfaces, nickel deposits tend to become streaky if stressed very highly, and double deposits of nickel and chromium should be used in those cases. Deposition on hardened or glass-hard surfaces should not be attempted without expert advice. The deposit can be machined similarly to steel but machining work-hardens the nickel. The strength of adhesion of nickel deposits to various metals is given in the following table, fracture taking place in the base metal in all cases.

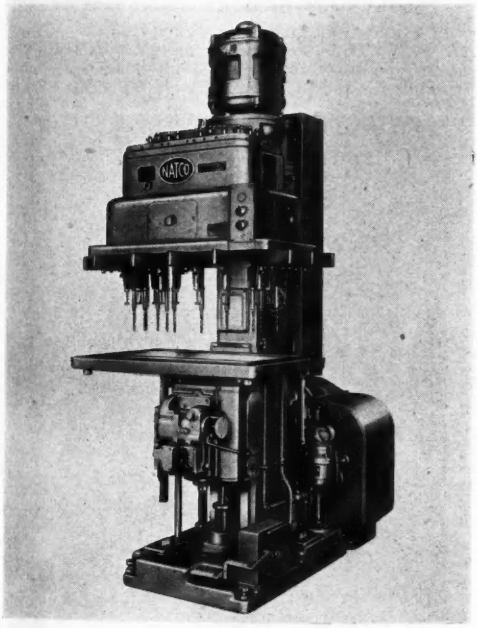
Base metal	Form	Strength of adhesion of deposit in psi
Mild steel	rolled bar	62,500
Nickel steel	forged	71,000
Copper	drawn rod	41,000
70/30 Brass	drawn rod	52,500
60/40 Brass	drawn rod	43,000
Aluminium	drawn rod	7,800
Tin	cast thin strip	4,000
Lead	deposit on steel	2,300

The base metal surface should be very smooth (fine-machined or ground) and grooves, notches and sharp edges should be rounded if faults in the deposit are to be avoided. The thickness of the deposit is not limited but the time required by the deposition process and its difficulties increase as the thickness of the deposit grows. Generally, a deposit of 0·1 inch is regarded as an economic limit. Very thin deposits, e.g. of 0·005 inch can be deposited, but, in general, it is found better to deposit a thicker layer and to machine it down to the required tolerances. External plane or cylindrical surfaces are easily covered with the deposit, but sharp edges and acute angles offer difficulties and thicker layers should be applied. Scratched and grooved surfaces, and holes of greater depth than width require special care if they cannot be smoothed out. Faults in the deposit are rare, although sometimes very small pits cannot be avoided. The temperature of the bath is 200° C, but if necessary, can be reduced to 100° C in order to avoid detriment to the heat treatment of the base metal. A sound adherent nickel deposit also, to some extent, restores the strength of a part weakened by faulty machining or wear. If, however, wear has work-hardened the base metal, other methods of building-up may be preferable. Special precautions must be taken for nickel deposition on alloy steels or non-ferrous or light alloys.

New Materials, Processes and Equipment

MULTI-DRILLERS AND TAPPERS

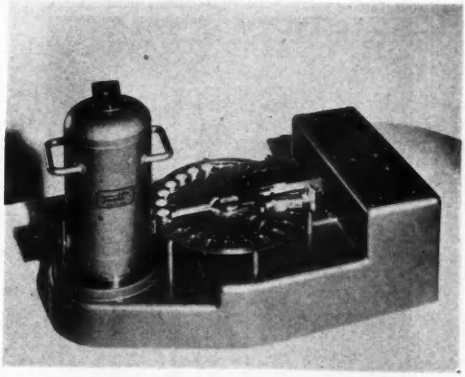
A new line of high speed sensitive multi-drillers and tappers has just been announced by the National Automatic Tool Co. of Richmond, Indiana. This new line consists of Heavy Duty and Light Duty Model H-5 and H-6 Multi-Spindle Drillers and Tappers which are of similar design, and vary only with regard to the size of the drilling area of the head, horse-power of head drive motor, number of spindles in the head and size of the work table. Incorporated in the design of these small NATCO machines are many improved and superior features. They will handle a wide range of small and medium sized work and allow multiple drilling or tapping of small holes with a minimum of cost.



These machines may be furnished as a Driller only, as a Tapper only, or as a combination Driller or Tapper. They are provided with change gears and a quick change speed mechanism for correct spindle speeds, a vertical adjustment on the spindles to compensate for the grinding of drills or the use of different length drills, and a feature in the design of the spindle proper permits float for tapping operations. The machines are equipped with a reversing motor drive which eliminates all belts and clutches for tapping operations.

AUTOMATIC SAMPLE CHANGER

Tracerlab Inc., Boston, announces the availability of the first commercially manufactured automatic sample changer for making radioactivity measurements. Laboratories which make routine radioassays on large numbers of samples will find that the Automatic Sample Changer permits more accurate and reproducible results, gives results faster, and makes substantial savings in the number of technicians required for radioassay work. It permits radioactivity measurements on as many as twenty-five samples to be made without attention of a technician and results are automatically printed on a paper tape. The Automatic Sample Changer can also be used for automatically making half life measurements on short half life materials or, with minor changes, can be used for obtaining energy spectrum data on a particular sample by automatically placing over the sample absorbers or filters of varying density and thickness and making measurements of the amount of activity detected by the Geiger-Müller tube. The photograph shows the Automatic Sample Changer with the Shielded Manual Sample Changer mounted on it. Connections for power and amplified G-M tube pulse are made to the Autoscaler with a single cable and a single cable and a second cable connects the Automatic Sample Changer with the Tracergraph.



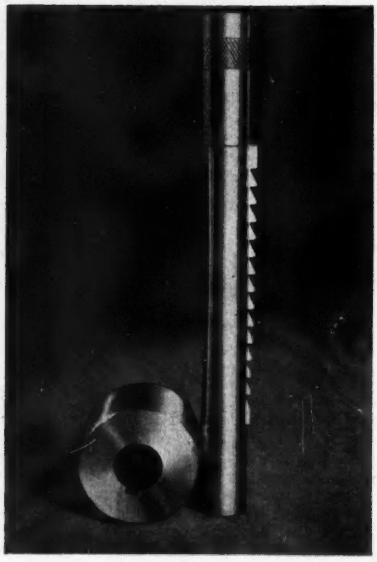
Twenty-five samples are loaded in slots on the circular disc. An arm with a ring on 1 inch inside diameter is dropped over the sample and pushes it into the Shielded Manual Sample Changer, stopping when it is directly centred under the G-M tube. When properly positioned, a relay starts the Autoscaler which has been previously set to indicate the number of counts consistent with the statistical accuracy desired. The Autoscaler will count to a predetermined number of counts and then shut itself off automatically. Beside each printed time interval is also printed the sample number for positive identification.

"SCOTCH-WELD" BONDING FILM

An unsupported film of pure adhesive that provides a metal-to-metal bond resistant to shear tests up to 3,500 psi was announced by Minnesota Mining and Manufacturing Co., St. Paul, Minn. It is transparent, roughly resembles cellophane in appearance, is provided in rolls like tape and is not tacky to the touch. It is 100 per cent adhesive, with no supporting material in the film. The film is placed between units to be bonded and is cured by simultaneous application of heat and pressure—a heat of 300 to 500° F for 5 to 60 minutes, and a pressure of 25 to 100 psi, both varying with the type of bond desired. Manufacturer's tests reveal shear strengths to 3,500 psi in bonding ·064-in aluminium Alclad to itself. In bonding the same metal to fibre, wood and plastic surfaces, the non-metallic materials delaminated at 880, 1,770 and 2,800 psi respectively. The adhesive did not fail.

THE HORTON KEY-WAY BROACH

This broach, made by The Sloan Machine Co., Cleveland 10, Ohio, will be of particular value to jobbing machine shops, builders of special machinery and repair and maintenance departments. It is a simple, sturdy push broach consisting of a hardened and ground body slotted for a high speed steel blade which is firmly held in place by a heavy locking nut.

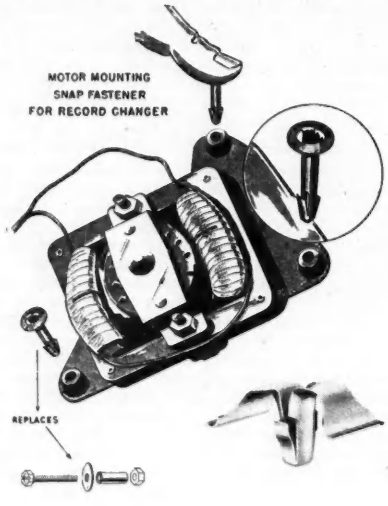


This broach can be used in any arbor press and every keyway made will be absolutely central with the hole, square and to the exact width and depth. The number of passes that must be made to complete the keyway will vary from two to five depending on the size of the bore.

PUSH-IN TYPE SNAP FASTENERS

Push-in type snap fasteners, developed by Shakeproof, Inc., Chicago, are stamped from spring steel and form a resilient, vibration-resistant lock to hold sheets, panels, brackets, etc., in place. Their chief advantages are reduced assembly time and material cost savings. In operation, a quick push with the thumb seats the fastener securely and no further rotation or setting operation is necessary. On many applications, because the fasteners eliminate the need for nuts, washers, spacers, etc., one snap fastener replaces four or even more separate pieces. The operating principles of the snap fasteners is very simple. In forming, extended spring prongs are stamped as an integral part of the fastener. These prongs compress as the fastener is pushed into a punched or drilled hole and spring out again on the far side of the joint to hold the assembly together under constant spring tension. In the composite photograph above, a snap fastener is shown as applied to shock mounting a small motor assembly. In this particular case, inserted through rubber vibration clamping grommets, the snap fasteners replace a screw, washer, spacer and nut. Assembly time is reduced by more than half.

Another type of Shakeproof snap fastener for locking metal sheets securely together is shown in the lower right hand corner.



SELF-CENTERING DRILL TURRET

This drill turret, made by Howe and Fant, Inc. of South Norwalk, Connecticut, under the trademark Lign-o-matic is so constructed that its six spindles are automatically centred and aligned by the drill press spindle.

The turret thus has a sustained accuracy equivalent to the drill press to which it is attached. The six Lign-o-matic spindles are mounted on specially designed floating bearings so that they are free to move laterally and angularly as well as to rotate. As the tapered driving unit, attached to the drill press spindle, engages the mating taper of a Lign-o-matic spindle, it centres and aligns the Lign-o-matic spindle and locks it in that position until the drilling operation is completed. This patented construction overcomes the fundamental difficulty of aligning with a live spindle which has heretofore prevented successful application of the turret principle to drilling machines. Another important feature is that Lign-o-matic has no gears, teeth, or projections of any kind to clash and wear. Driving power is transmitted directly from machine spindle to Lign-o-matic by the same tapered surfaces that accomplish alignment.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

BEARINGS AND BEARING METALS

Wear Factor for Transmission and Engine Bearings

By F. PICARD and L. BUTY. (From *Machines et Métaux, France*, Vol. 32, No. 357, May, 1948, pp. 157-160, 3 illustrations.)

THE wear factor characterises mechanical running conditions of a bearing under a definite bearing pressure and at a definite rubbing speed. It permits comparison of different anti-friction alloys and determines where each can best be employed. It also gives the basic data for the dimensional design of bearings. In a first, simplified, approach, the energy dissipated in heat due to bearing friction has been assumed to be proportional to the product $p \times V$, where p is the bearing pressure per sq. in. of projected bearing area and V the rubbing speed. Experience shows, however, that the influence of the speed is more pronounced than that of the pressure, and it can be shown from the theory of perfect film lubrication that the frictional energy must be proportional to the expression $\sqrt{p_m}\, V^2$. (See also *The Engineers' Digest*, February, 1948, p. 53). This expression conforms better to experience and has been adopted by the French Bearings Research Centre for further investigation. However, only the mean pressure p_m appears in this factor, although it is obvious that pressure variations must also determine wear conditions, if only by their influence on lubrication.

A table of values for the wear factor of normal radial and thrust bearings of rotating machinery is given. In reciprocating engines, pressure and rubbing speed vary to a marked extent during each stroke, and mean values must first be approximately calculated in accordance with formulae given in *The Engineers' Digest* of February, 1948, p. 53. These values then permit of establishing a mean wear factor for reciprocating machinery. For a comparison of different types of engines, running conditions must finally be standardised as described in the article previously mentioned. A second table gives normal values of the wear factor for automobile, lorry and marine engines. The factor is lowest for slow-running marine engines, higher for automobiles at medium touring speed and maximum for lorries and traction vehicles. If it exceeds a certain value, namely, 250, p being taken as kg/cm² and V as m/sec, bearings with normal anti-friction bearing metals appear to give rise to frequent trouble. The Bearings Research Centre aims to establish the best running conditions for the various bearing metals and issues data sheets to be completed by engine users to enable it to draw conclusions from the widest possible field of experience.

ELECTRIC BATTERIES

Storage Battery Addition Agents

By EUGENE WILLIHNGANZ. (From a paper of *The Electrochemical Society, U.S.A.*, Preprint 92-32, October, 1947, pp. 385-406, 14 illustrations.)

IN the manufacture of storage batteries it is customary to include a number of addition agents, known collectively as an expander, in the negative plate. This expander consists of a mixture of lampblack, barium sulfate, and an organic material usually derived from the lignin fraction of wood. This expander greatly improves the capacity of the battery.

The discharge characteristics of lead storage batteries have been studied to learn why this expander improves the capacity. It is shown that it renders porous the lead sulfate formed during discharge. The organic ingredient of the expander prevents lead sulfate, formed by discharge, from coating the underlying lead metal and blocking off the surface. Barium sulfate contributes to this action by offering an alternate place for lead sulfate to crystallize. These actions are responsible for the observed increase in the capacity of storage batteries.

FOUNDRY PRACTICE

Graphical Representation of a Sand Fineness Test

By E. O. LISSELL. (From *Gjuteriet, Sweden*, Vol. 38, No. 3, March, 1948, pp. 43-51, 7 illustrations.)

FINENESS has a marked influence on the properties of foundry sands. The test for determining the fineness of such sands is, therefore, important, as is also the graphical representation of the sizing analysis. Filling the amount of sand retained on each sieve in the AFA set of sieves, as well as the pan material, directly into test tubes, or drawing a direct block diagram to illustrate the result of the analysis, as is often seen in literature, is wrong. The intervals between number 6, 12 and 20 sieves as well as between sieve 270 and the upper limit for the AFA clay are not comparable to the other intervals for which the logarithm of the larger opening minus the logarithm of the smaller one is constant. The amounts retained on the 20, 12 and 6 sieves should be halved and the pan material multiplied by 0·35 before the block diagram is plotted or the test tubes are filled. This is actually what is done when the relative or logarithmic frequency curve is used. The logarithmic frequency curve is recommended as the best means of representing the sieve analysis. However, when several different sands are to be compared on the same diagram, the logarithmic cumulative curve is generally to be preferred. A standard diagram for logarithmic plots which takes in both the American and German sieve sets is described.

HEAT TREATMENT

Hardenability Specifications

By H. B. KNOWLTON. (From *The Iron Age, U.S.A.*, Vol. 161, No. 23, June 3, 1948, pp. 72-77 and p. 177, 4 illustrations.)

WITH the steel user urging closer control over hardenability and the steelmaker finding it increasingly difficult to meet chemical analysis specifications, it appears that there is much to be said, both metallurgically and economically, in favour of purchasing alloy steels under controlled hardenability specifications. Data are presented in this article indicating the practical significance of minimum hardenability in regulating physical properties, and also the relation of minimum hardenability to per cent of martensite. The author also discusses the effect of tempering on hardenability and points to the desirability of performing the measurement of this relationship in the hardenability test specimens.

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KINEMATICS

Analyzing Circular-Arc Cams

By P. K. SLAYMAKER. (From *Machine Design, U.S.A.*, Vol. 20, No. 7, July, 1948, pp. 129-131, 4 illustrations.)

DISK cams with profiles made up of circular arcs offer a popular method of obtaining controlled motion. The sides or flanks of the cam are arcs drawn tangent to the base circle and to the rounded nose. Layout of such cams is easy and convenient, but it is not generally realized that there exists an extremely simple method of analyzing the follower motion when a flat-faced follower is used. The purpose of the present article is to explain this construction, which affords the designer a quick method of determining velocities and accelerations needed in designing the complete cam and follower mechanism.

METALLURGICAL ENGINEERING

A Magnetic Method for Microscopic Examination

By S. BERGH. (From *Jernkontorets Annaler, Sweden*, Vol. 132, No. 4, April, 1948, pp. 105-109, 6 illustrations.)

IN order to make ferromagnetic constituents in the structure of austenitic steels visible under the microscope AKIMOFF used a suspension of magnetic powder, in which the specimen was immersed during examination. This method is to some extent inconvenient. To overcome this, the author proposes another method, using a suspension of magnetite powder in a solution of high specific gravity (Clerici's solution: 2 parts of formiate of thallium, 1 part of malonate of thallium), and applying a thin film of this suspension under a glass cover to the specimen. A bar magnet is fixed to the specimen with plasticine. By this method it is possible to use higher magnifications, the image is not disturbed by successive sedimentation of magnetic powder, and the work is simplified.

NON-FERROUS METALS

The Measurement of the Magnetic Properties of Non-Ferrous Metals

By ZD. TRNKA. (From *Elektrotechnicky Obzor, Czechoslovakia*, Vol. 36, No. 16, August 27, 1947, pp. 305-306, 2 illustrations.)

IMPURITIES may considerably increase the magnetic susceptibility of non-ferrous metals. Such metals, when used in galvanometers, compasses, etc., will introduce errors. To determine the susceptibility of these metals, small samples in rod form are prepared. By approaching the sample to a sensitive magnetometer, we obtain a qualitative indication of the magnetic properties.

An accurate method of determining the susceptibility of the rod makes use of the instrument shown in Fig. 1. The rod t with a plane mirror z attached is magnetized by the homogeneous field of the permanent magnets $p - p'$ and suspended by Z exerting torsional control. A coil C is excited by current of a finely regulated frequency which approaches the natural frequency of the rod-mirror system. The field of the coil causes oscillation of the rod and the frequency of the current in coil C is set for maximum deflection. This maximum deflection is proportional to the susceptibility of the sample rod. A note is made of the current I_1 in the coil which resulted in this deflection. A rod of known susceptibility κ_2 and equal moment of inertia is then placed in the instrument, and note is also taken of the current I_2 in the coil which causes the same maximum deflection as before. The unknown susceptibility is then

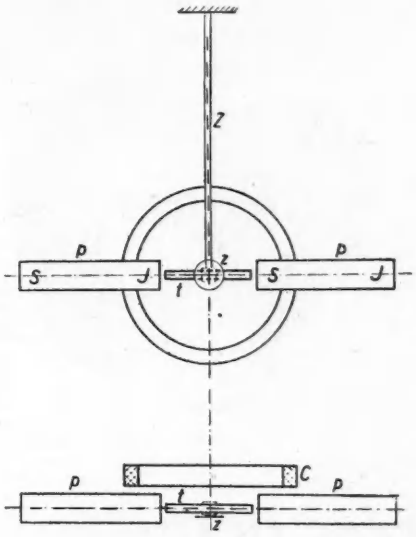

Fig. 1.

$$\kappa_1 = \kappa_2 \frac{I_2}{I_1}$$

Susceptibilities even of the order of 10^{-4} can be measured in this way. The rods of known properties had a permeability of $\mu_2 = 10$ and the corresponding $\kappa_2 = \dfrac{\mu_2 - 1}{4\pi}$ has been calculated.

WEAR TECHNOLOGY

Wear Technology

By H. WAHL. (From *Die Technik, Germany*, Vol. 3, No. 5, May, 1948, pp. 193-204, 24 illustrations).

A COMPREHENSIVE survey is made of some general problems of practical wear technology. Wear is defined as the gradual change of the surface of solid bodies, especially by the mechanical loosening of small particles of the material in question. The economical importance of wear reduction is stressed and an analysis is made of the present state of wear research. Preliminary instructions are given for the solution of practical wear problems. Three principal rules are mentioned, viz.:—(a) the greatest possible reduction of "wear attack," (b) the greatest possible increase of wear resistance, and (c) the greatest possible reduction of the harmful effects of wear still continuing, as usually the total elimination of wear cannot be obtained economically. The wear resistance is concentrated in the basic moving body. The wear attack is produced by the combination of a counterbody, which may be solid, liquid or gaseous, an intermediate body, e.g., lubricant, the conditions of movement, the active forces and their duration. The manner in which these different components may be influenced is discussed in detail.

A number of graphs and tables are given, analyzing the various kinds of wear, suitable practical means for reducing it, the use of various machines for wear tests, the influence of materials, speed, pressure, etc.

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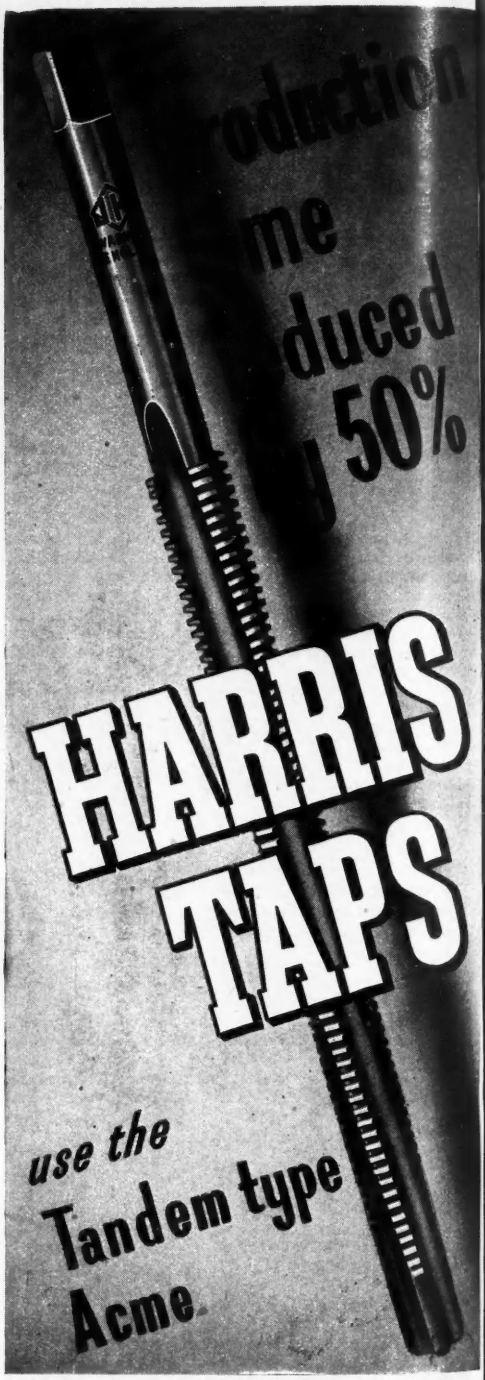

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NEWS OF THE MONTH

PERSONAL

Mr. Leonard Austin, M.I.P.E., has been appointed joint managing director of Wadkin Limited, Leicester.

Mr. F. H. Boud has been appointed assistant power station superintendent, Bedford, **Mr. L. P. Float,** assistant power station superintendent, Watford, and **Mr. N. Neal,** communications engineer at Divisional Headquarters of The British Electricity Authority, Eastern Division.

Mr. R. E. C. Cadman has been appointed general manager of the Wakefield works of British Ropes Ltd.

Captain A. W. Clarke, C.B.E., D.S.O., R.N. (ret.), has been appointed Chief of Naval Information.

Mr. Arthur Coe has been appointed director of the Combustion Appliance Makers' Association.

Mr. H. Constant, M.A., has been appointed director of the Ministry of Supply's National Gas Turbine Establishment in succession to Dr. H. Roxbee Cox who has been appointed Chief Scientist at the Ministry of Fuel and Power.

Mr. S. H. Fisher has been appointed deputy chief regional officer, London Midland Region of The Railway Executive.

Mr. L. E. Ford, O.B.E., has been appointed general manager of the Port of London Authority, Tower Hill, London, E.C.3.

Mr. E. A. Foster-Davis, of 104 Tower Street, P.O. Box 317, Kingston, Jamaica, has been appointed direct factory representative of Specialloid Ltd. in Jamaica.

Mr. G. C. Handley has been appointed industrial sales manager of Worthington-Simpson Ltd.

Mr. Edmund L. Hann has relinquished his office of managing director of Powell Duffryn Ltd., but consented to remain chairman of the Company. **Mr. H. H. Merrett,** managing director, has been appointed deputy chairman.

Mr. G. W. Hand and **Mr. T. H. Burleigh** have been appointed directors of Firth Brown Tools Ltd., Sheffield.

Mr. C. H. Harris has been appointed London sales manager of A. A. Jones & Shipman Ltd., who have opened their London office at Buckingham House, Buckingham Street, Adelphi, W.C.2.

Mr. G. A. Heslip, M.I.Mech.E., M.S.A.E., has been appointed production manager of the Climax Rock Drill and Engineering Co., Ltd., Carn Brea, Cornwall.

Mr. John H. D. Hobson, foundry manager of Tees Side Bridge and Engineering Works Ltd., has been appointed general manager of F. Issels & Son Ltd., Bulawayo, an associated company of John Brown & Co., Ltd.

Mr. B. E. G. Odds and **Mr. Fitzherbert Wright** have been appointed joint deputy managing directors of Aveling-Barford Ltd., and **Mr. E. R. Howlett** and **Mr. C. J. Ritchie** have been elected directors.

Mr. D. A. Palmer has been elected chairman of Reavell & Co., Ltd., Ranelagh Works, Ipswich, and Mr. R. L. Quertier, deputy chairman. **Mr. Quertier** and **Lieut.-Colonel K. Reavell** have been appointed joint managing directors.

Viscount Ridley has been elected to the board of Head, Wrightson & Co., Ltd.

Mr. Gerald Steel has been nominated vice-president of the Iron and Steel Institute, and **Mr. W. Barr** has become a member of the Council. **Mr. A. H. Goodger** will serve as an honorary member of the Council during his presidency of the Manchester Metallurgical Society.

Mr. H. Trowsdale has been appointed station superintendent of the Falkirk Power Station, South-East Scotland Division, British Electricity Authority.

Northern Ireland Transport Authority. The Northern Ireland Minister of Commerce has appointed the following to be members of the new Northern Ireland Transport Authority:—Major F. A. Pope (Chairman), Mr. W. A. Edmenson, Mr. R. P. C. Gotto, Sir Dudley E. B. McCorkell, Mr. S. J. McCoubrey, Lt.-Col. Viscount Massereene and Ferrard, Major W. G. Nixon, Mr. J. S. Rogers, Mr. D. Wilson Smyth and Mr. A. E. Swain.

The Authority is responsible for co-ordinating road and rail transport.

The **Ministry of Fuel and Power** announces that, in consequence of the winding up of the Electricity Commission, the following appointments have been made in the Electricity Division of the Ministry at Cromwell House, Dean Stanley Street, London, S.W.1:—Mr. A. N. East, chief engineering inspector; Mr. H. W. Grimmitt, deputy chief engineering inspector; Mr. A. H. Linton, engineering inspector; and Mr. S. Howarth, chief meter examiner.

Smoke Abatement Conference. The Minister of Fuel and Power, Mr. Hugh Gaitskell, is to attend and will address the annual conference of the National Smoke Abatement Society, which is to be held at Cheltenham on 30th September and 1st October. His presence will emphasize the growing alliance between the movement for smoke prevention and the national drive for fuel efficiency. 400 delegates from local authorities, Government departments, fuel and other technical organisations, as well as individual members, will be present.

Standard Locomotive Works Practice. Behind the announcement by the Locomotive Manufacturers' Association of Great Britain that the whole locomotive industry in that country has adopted a single comprehensive standard works practice lies one of the biggest opportunities ever present for improving in a practical sense locomotive building, running, and repair.

All locomotives—steam, diesel, electric, and from main-line to industrial duties—built in Great Britain for export will henceforth be constructed to standard tolerances to provide interchangeability and standardisation to such a degree as to simplify considerably running-shed maintenance replacements, shop overhauls, and the ordering and assembly of spare parts, particularly where a user has locomotives from two or three makers.

The only exceptions to this rule will be those where customers specifically ask for other tolerances to be embodied. Actually, the limits and fits now put forward do not cut across the majority of existing individual specifications, all of which were carefully considered by the L.M.A. sub-committee charged with the work.

Built up partly on the "hole basis" and partly on the "shaft basis" to meet the special problems of locomotive parts manufacture and renewal, the standard practice now adopted is listed in the Association's book under 14 headings, covering all of the machinery and chassis.

There seems to be no weighty reason why practices now adopted after lengthy investigation by plants capable of supplying over 2,000 locomotives a year throughout the world could not be introduced as a world-wide standard, and eliminating the multifarious systems of small application which help to make locomotive manufacture and locomotive maintenance more costly and difficult than they need be.

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BUSINESS NOTES

The Hoffmann Manufacturing Co., Ltd., Chelmsford, Essex, has opened a new branch office at 48-50 Blackness Road, Dundee, Telephone 3265. An emergency stock of Hoffmann Ball and Roller Bearings will be carried, and a competent bearing engineer will provide for engineering service.

British Insulated Callender's Cables Ltd. has moved its Edinburgh branch office to 80-82 Hannover Street, Edinburgh.

The Leytonstone Jig and Tool Co., Ltd., has opened an office at 31 Willcocks Street, Toronto 5, Canada. Air Vice-Marshall D. F. Stevenson, C.B., is the resident director.

The Export Licensing Branch of The Board of Trade has moved its Headquarters to Regis House, King William Street, London, E.C.4. Telephone Avenue 3111.

Macrome Ltd. has appointed the following Agents :—West African Engineering Co., Ltd., P.O. Box 8, Secondi, Gold Coast Colony ; The British Italian Trading Co., Ltd., 75 Bishopsgate, London, E.C.2 ; and Nielson & Danielsen A/S., Tollbodgt. 25, Oslo, Norway.

F. A. Hughes & Co., Ltd., have removed their offices to Bath House, 82 Piccadilly, London, W.1. Telephone Grosvenor 6300.

A. R. Parsons & Co. have transferred their Head Office and Works from 113-119 Regents Park Road, London, N.W.1. to their London Colney factory. The new address is Fidelater Works, 275 Main Road, London Colney, Herts. Tel. : London Colney 3113/14.

BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 24 Victoria Street, London, S.W.1.)

The Importance of Correct Boiler Water Treatment (B.S. 1170). Attention is again called to the fact that the British Standards Institution has recently published (December, 1947) a Code of Practice for the Treatment of Water for Marine Boilers, with a section on Engineering Factors, which should be of the greatest value to staffs of all sea-going Chief Engineers and Marine Engineering Superintendents. Price 10/6.

Moulded Insulating Materials for General Electric Purposes. A series of British Standards dealing with the performance of moulded materials used for electrical insulation is approaching completion. The first of these specifications—B.S.488, which was first published in 1933—has been revised and is now available as B.S. 488, 1948. This series of standards approaches the problems of specifying electrical insulating material from the point of view of the designer of electrical equipment, who requires certain properties to meet given conditions of use, but desires also a free choice among the various materials that might meet these needs. Each of the specifications in the series is therefore designed to include all the types of materials that meet the requirements of the branch of electrical work covered by the specification, and the whole series provides similar methods of test for all types of materials where this is possible, to permit comparison of technical data.

The revision of B.S. 488 covers moulded materials (excluding ceramics) for use as insulating materials for general electric purposes on currents up to 250 volts, and at working temperatures not exceeding 100° C.

The standard provides for four grades of moulded materials based on resistance to heat and specifying qualifying limits for electrical properties, resistance to heat, strength and general properties for each grade. Price 2/6.

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CATALOGUES RECEIVED

A. C. Wickman Ltd., Coventry, sole agents for the High Speed Die Piercing and Polishing Machine produced by Rudkin & Riley Ltd., Aylestone, Leicester, have issued a new catalogue with description of this machine.

Metway Electrical Industries Ltd., King Street, Brighton 1, has published a 12-page brochure and price list of their P.V.C. Armoured Flexible Conduit Tubing. Applications include protection of flexible leads to electric motors and machine tools, travelling cranes, wiring on motor vehicles and various domestic appliances. P.V.C. tubings are, however, not suitable for use with gas or steam.

William Geipel Ltd., 156-170 Bermondsey Street, London, S.E.1, published a folder on their range of standard electric control gear.

The Mond Nickel Co., Ltd., Grosvenor House, London, W.1, issued a booklet on "Nickel Alloy Steels—A Summary of their Properties and Applications." It reviews briefly the range of nickel alloy steels in the light of the effect of nickel and other alloying elements, and includes typical properties and applications of normalised, case-hardened, and fully treated steels.

H. Brammer & Co. Ltd., East Street Mills, Leeds 9, published a well designed and profoundly illustrated 32-page catalogue on their detachable V-Link Belting which was invented and patented throughout the world by Mr. H. Brammer, the managing director of the Company. Numerous diagrams, photographic illustrations and h.p. rating tables make the catalogue of great usefulness to design engineers and to all interested in power transmission.

BRITISH EXHIBITION IN COPENHAGEN.

More than 1000 British manufacturers will display their products from September 18 to October 3 at the largest and most comprehensive display of British goods ever held in Denmark or the Scandinavian countries.

The Exhibition will be opened on September 18 by the King of Denmark in the presence of the Duke and Duchess of Gloucester. Five central buildings, the Forum, the Tivoli Gardens, the Royal Riding House, the Magasin du Nord, and the Gutenberghus, will be used as Exhibition Centres.

A representative assembly of British engineering equipment including diesel engines, machine tools, machinery, components, measuring equipment, etc., will be displayed.

The following firms have sent us details of their exhibits :—

The British Thomson-Houston Co. Ltd.'s display includes horizontal and vertical variable speed A.C. commutator motors, ¼ H.P. die-cast frame motors, fractional horsepower motors of various types, thrustors, suds pumps, electronic apparatus, including a magnetron as used in B.T.H. marine radar equipment. The stand is illuminated by B.T.H. fluorescent equipment.

The General Electric Co. Ltd. shows an attractive three-dimensional picture of the Woolwich Power Station, a model of a Sherwen vibrating screen, representative of the mining and materials handling plant manufactured by the Company's Fraser & Chalmers Engineering Works, various types of infra-red heating equipment, electronic measuring instruments, a comprehensive selection of lighting and control equipment, ventilating equipment manufactured by Woods of Colchester, an associate company of G.E.C., including aerofoil fans, man cooler and propeller fans.

The Goodyear Tyre & Rubber Co. shows the working model of the world's highest industrial "lift," the actual installation of which lifts 1000 tons of coal an hour over a height of 715 feet on a single flight conveyor belt.

Marconi Instruments Ltd. will display in the Scientific Instrument Manufacturers' Association Group exhibit Communications Test Gear, Industrial Measuring Instruments, Electro-Medical Apparatus and the Marconi 200 mA and 500 mA X-Ray Equipment. The exhibits include equipment of very latest design.

Tangyes Ltd. will show a variety of products, amongst them their new type 25 ton Universal Testing Machine and Hydraulic Lifting Jacks of entirely new design.

Exhibition of Electronic Instruments. A new range of instruments for use in industry and research will be on display at the Grand Hotel, Birmingham, from September 20th to 24th. The exhibition will be open from 10 a.m. to 6 p.m. daily and admission which is free will be limited to ticket holders only. Further information and tickets will be supplied to the trade only on application to :—Cinema-Television Limited, Worsley Bridge Road, Lower Sydenham, London, S.E.26.

One of the interesting exhibits will be the Industrial Electronic Metal Detector. Now for the first time in this country an electronic metal detector is available to detect tramp metal in raw material and in manufactured goods before they leave the factories and so help to maintain the traditionally high standard of British goods.

The applications of another type of metal detector on display are to trace buried pipes, to locate lost man-hole covers and to test timber to ensure that the wood contains no hidden metal when it is used by the saw mill. It is also used to detect the presence of metal in beasts.

An Electronic Counter will be shown, which, counting at speeds of up to 30,000 per minute can record the output of machines, or can be used in research work to count radiation particles. Another type of counter also shown is the Decimal Counter Chronometer, which instrument will measure short time intervals of up to one second in duration with an accuracy of ± 10 microseconds. Its applications are to be found in ballistics and velocity measurements, counting, accurate timing, etc. One particularly interesting application is in frequency measurement, where, when a control button is depressed the unknown frequency is measured and directly indicated to five significant figures.

In addition three new oscilloscopes will be demonstrated.

The Institution of Production Engineers. On September 1st, 1950, the examination requirements of the Institution will undergo considerable alteration and an associate membership examination will be introduced. A 40-page booklet, just issued, gives full details of the examination regulations and membership qualifications. The new examination will conform closely with existing National Certificate Schemes in Technical Colleges. The booklet includes also the examination syllabuses, particulars of subscriptions and examination fees.

Archdale Machine Tools, 1868-1948. Frank, James, Fred and Will Archdale, the present directors of James Archdale & Co. Ltd., of Ledsam Street, Birmingham, and Blackpole, Worcester, commemorated the eightieth anniversary of the Company by publishing a 100-page book devoted to the history of one of the largest and most progressive enterprises of the British machine tool industry.

Son of a miner in the West Riding of Yorkshire, James Archdale was sent out to work at the tender age of nine. He went at first into the mine, served later his apprenticeship with the Leeds engineering firms Kitson & Co., Benjamin Walker, John Fowler & Co. and Greenwood & Batley, and spent two years as foreman of the machine tool shop of Tangyes Ltd., Birmingham, before starting in business on his own account in 1868. The experience gained with these firms, his "flair" for mechanical engineering and his personal qualities enabled James Archdale to lay sound foundation to the business which has been developed to the present manufacturing organization with a world-wide reputation.

The book is well illustrated with numerous drawings and photographs of the machine tools made by James Archdale & Co. during the past eighty years and it makes most interesting and fascinating reading.

Mullard Readership in Electronics. The Mullard Company's recognition of the necessity for industrial organisations to accept some measure of responsibility, and to assist actively in the problems of the future education of electrical engineers, has prompted them to take practical steps in this connection by offering to finance a Readership in Electronics at the City and Guilds College of Imperial College.

Following discussions with Professor Willis Jackson on this subject an approach was made to the Rector of Imperial College, Sir Richard Southwell, outlining the Company's views and desires, and this has resulted in the authorities of the University of London accepting the offer made by the Mullard Company.

The Readership will be mainly concerned with post-graduate teaching of research.

International Technical Congress in Cairo. Following an invitation extended by the Egyptian Government, the second International Technical Congress will be held in Cairo from March 20th to 26th, 1949. Programme of the Congress which will take place at the same time as the Egyptian Agricultural and Industrial Exhibition will include discussions on Industrial Raw Materials and their rational Utilization throughout the World; the social aspect of technical development and of raw material problems; and the Problem of Water in the Middle East. The Cairo Reception Committee of the Congress will be presided over by H. E. Hussein Bey, chairman of the Committee of Public Works of the Egyptian Parliament.

Society of Inventors. The second annual exhibition will be held at the Chamber of Commerce Buildings, Birmingham, from 25th to 30th October. Entries will be received from members of the Society as well as from non-members. Further information can be obtained from Mr. Jas. E. Wilson, Trent Valley House, Lichfield, Staffs.

The **North of England Ironfounders Association,** whose members constitute a powerful industrial group in North-East England, has affiliated to the Engineering Industries Association.

Our technical representatives will gladly advise you on the varying types of the Weston Seals. These include our Gitseals and Weston Oil Retainers fitted with best quality chrome leather inserts, also the Weston Synthetic Rubber Oilseals all precision made to give the lightest possible contact consistent with efficient sealing.

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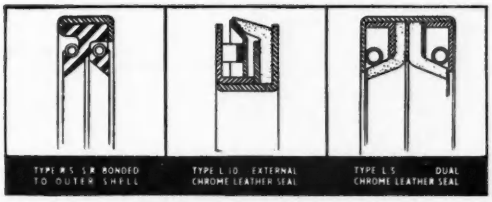

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BOOKS

Factory Administration in Practice. By W. J. Hiscox and James Stirling. Publishers: Sir Isaac Pitman & Sons Ltd., Parker Street, Kingsway, London, W.C.2. 6th Edition. 300 pp. 98 figures. Price 8/6.

This thoroughly practical book on factory administration and methods of organization offers a comprehensive course of great importance to works managers and others interested in the efficient administration of industrial enterprises. In 38 chapters the book deals with all phases of factory administration, including purchasing and sales departments, stores, labour administration and methods of control.

The book is also recommended to students preparing for professional examinations of the Institute of Cost and Works Accountants.

The Design of Welded Steel Structures. By A. Ramsay Moon. Publishers: Sir Isaac Pitman & Sons Ltd., Parker Street, Kingsway, London, W.C.2. 134 pp. 102 figures. Price 18/-.

This extremely useful little textbook will be of much value to design engineers who want to have at easy reach design data for arc welded structures. The L.C.C. Welding Regulations are included in the book as appendix.

Automobile Chassis Design. By R. Dean-Averns. Publishers: Iliffe & Sons Ltd., Dorset House, London, S.E.1. 238 pages, 160 diagrams and photographs. Price 15/-.

There are many good books on automobile motors, but few up-to-date books on the fundamentals and technique involved in the design of the automobile chassis.

The present book fills this gap, and as it is written by a competent author on this subject it meets a definite need. Much of the subject matter is evolved in connection with heavy-design, but, being of a general character, it can be applied to light car design.

The chapters include: Vehicle Performance, Front Axles and Suspension, Vehicle Braking and Performance, The Frame, Road Springs (including shock absorbers and the German stabilus method). The Transmission, Rear Axles, The Differential, The Chassis and appendix on the Chief Statutory Requirements Affecting Motor Vehicles.

Thoroughly to be recommended.

Diesel Operation and Maintenance. By Orville L. Adams. Publishers: Prentice-Hall, Inc., New York and Chapman & Hall, Ltd., 37 Essex Street, London, W.C.2. 366 pages, numerous figures and tables. Price 28/- net.

This is a practical book for those concerned with the operation and maintenance of Diesel engines. It is based on American experience, mostly with the U.S. Navy, but as it deals with fundamentals it should be of general interest.

The first five chapters explain the basic principles of engine operation, and the problems are presented in their relation to maintenance.

Another five chapters cover maintenance and repair procedures and the diagnosis of all engine difficulties with reference to failures of engine parts, fuels, combustion and lubrication.

The material presented is suitable for home-study and classroom instruction.

Arbeitsverfahren und Betriebsmittel. By Ing. Oskar Gönner. Publishers: Carl Hanser Verlag, München 27. 175 pages. 244 Figures. First Edition, Marks 8.20.

This book on Work Methods and Tools is devoted to the much neglected subject in literature, the production of pipe lines and pipe-like hollow parts.

Both text and the numerous illustrations are excellent and show the use of the most modern methods as applied to tube manufacture, tube bending, pipe flaring, rolling-in, etc. Both seamless and welded tubes are discussed, and special attention is paid to the thin-walled tube.

A generally useful book of the greatest interest for the aircraft industry.

The Fracture of Mild Steel Plate. Report No. 3 by C. F. Elam Tipper, M.A., D.Sc. Publishers: His Majesty's Stationery Office, Kingsway, London, W.C.2. Numerous plates, figures and diagrams. Price: 6s 6d.

Examinations of the causes of brittleness of mild steel used in shipbuilding was carried out in the Engineering Department, Cambridge University. The experimental results were submitted to the Admiralty Ship-Welding Sub-Committee. The reports have now been summarized in the following sections:—1. Description of fractures in actual ship plate. 2. Experiments designed to reproduce similar fractures in the laboratory. 3. Development of a notch test and study of the conditions of test as applied to ship plate. 4. Metallurgical investigation of the plates. 5. Fundamental experiments on the effect of notches on the ductility and fractures of mild steel. 6. Discussion of results in relation to the causes of brittleness in mild steel. 7. Tables. Data relating to the source, composition, mechanical and physical properties of the plates.

The Nickel Bulletin, Volume 21, No. 7. The article in the July issue "An Impact Test for Cast Iron" is of particular interest on account of the improvements in the quality of modern cast irons.

For the past 25 years, indeed until quite recently, cast irons have been regarded as brittle materials. With the improvements in quality introduced during the war years, the need for a test that will determine the differences in impact strength becomes apparent. The article contains data leading up to the establishment of a British Standard Test.

Abstracts in this issue of the Bulletin cover a wide range of subjects and include a United States summary of data on high temperature alloys, with an accompanying table of compositions in addition to the British Standard Specification for grey cast iron.

The Nickel Bulletin is obtainable, free of charge, from The Mond Nickel Company, Limited, Grosvenor House, Park Lane, London, W.1.

Classified Advertisements.

The Rate for all classified advertisements is 6d. per word; in bold print 9d. per word; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

SITUATIONS VACANT

ADMIRALTY (C.E. in C. Department). Vacancies exist for Architectural and Civil Engineering Assistants in the Drawing Office at H.M. Dockyards and Establishments at Home and Abroad.

Applicants must be competent draughtsmen and designers with experience in some of the following types of Building and Civil Engineering Structures and Works:—Structural Steelwork, Reinforced Concrete, Dock and Harbour Works, Accommodation, Residential, Office, Store and Factory Buildings, Oil Fuel Tanks and Pipe Lines, Airfield Construction and Maintenance, Rail, Road, Water, Sewerage and Heating services, etc.

The salary for Architectural and Civil Engineering Assistants is £300-£525 (consolidated London rate) subject to Provincial differentiation of £20 at the minimum and £30 at the maximum, with a weight for age allowance above or below the age of 21 up to a total addition of seven increases of £20 each on the London rate. Appropriate rates of Colonial Allowance are payable in addition at Foreign Stations.

Vacancies exist at present at: Pinner, Broughton Moor, Devonport, Glasgow, Liverpool, Milford Haven, Arbroath, Orkney, Invergordon, Fareham, Sheerness, Shotley, Ceylon Gibraltar, Malta, and Singapore.

The appointments are temporary.

Applications should be made, giving full particulars of experience, to The Civil Engineer-in-Chief, Admiralty, Chamberlain Way, Pinner, Middlesex.

WORK WANTED

CASTINGS.—A small amount of immediate available capacity for precision aluminium gravity die castings, zinc pressure die castings.—Write or 'phone **Thos. Ashworth & Co., Ltd.** (Dept. A/15/9), Vulcan Works, Burnley, Lancs. (Burnley 3505).

WANTED.—2-ton Hounsfield Tensometer or similar testing machine. State price, condition where available for inspection.—Box J.L.4, "The Engineers' Digest," 120, Wigmore Street, London, W.1.

MACHINERY, ETC., FOR SALE

HOW MUCH MOISTURE in foundry sands, ores, other raw materials? **"Speedy" Moisture Tester** gives the accurate answer in three minutes. Strongly built for use by unskilled labour. Portable; no electricity; inexpensive. Immediate dispatch from stock.—Write for Lab. Test Reports and illustrated Brochure to **Thos. Ashworth & Co., Ltd.** (Dept. ED/5), Burnley, Lancs. Agencies open.

WE HAVE LARGE STOCKS of Electrical Conduit and Fittings available for immediate disposal.—Box No. 83, "The Engineers' Digest," 120 Wigmore Street, London, W.1.

NISSEN TYPE HUTS, ex-Government stock, 36 ft. by 16ft., £70 and £80; 24 ft. by 16 ft., £56 and £45. Other buildings also available. Write for details.—Dept. 50, **Universal Supplies (Belvedere) Ltd.,** Crabtree Manorway. Belvedere, Kent. Telephone: ERITH 2948.

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